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HEAT TRANSFER PERFORMANCE OF A ROTATING HEAT PIPE USING DIFFERE--ETC(U)

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# THESIS

HEAT TRANSFER PERFORMANCE OF A
ROTATING HEAT PIPE
USING DIFFERENT CONDENSERS
AND WORKING FLUIDS

by

Hans-Joachim Weigel

December 1979

Thesis Advisor: P.J. Marto



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and the cooling water inlet. The main objective was to compare the different condenser configurations and working fluids with respect to the heat transfer achieved.

The best heat transfer performance was achieved by using the spiral Noranda tube and distilled water as working fluid. With this tube rotating at 2800 RPM the heat pipe was capable of transferring 6.5 KW at a vapor temperature of 90°C. The maximum increase in performance of 300% over the smooth cylinder occurred when using the Turbotec tube and Freon 113 as working fluid.

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# Heat Transfer Performance of a Rotating Heat Pipe Using Different Condensers and Working Fluids

by

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Submitted in partial fulfillment of the requirements for the degree of

MASTER OF SCIENCE IN ENGINEERING SCIENCE

from the

NAVAL POSTGRADUATE SCHOOL
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# ABSTRACT

A rotating heat pipe was tested using five different copper condensers (a smooth cylinder, a spiral Noranda tube, two Hitachi tubes and a Turbotec tube) and three different working fluids (distilled water, ethanol and Freon 113). All condensers were tested with filmwise condensation at rotational speeds of 700, 1400, 2800 RPM. The heat transfer rate of each run was measured and plotted against the temperature difference between the vapor and the cooling water inlet. The main objective was to compare the different condenser configurations and working fluids with respect to the heat transfer achieved.

The best heat transfer performance was achieved by using the spiral Noranda tube and distilled water as working fluid. With this tube rotating at 2800 RPM the heat pipe was capable of transferring 6.5 KW at a vapor temperature of 90$^{\circ}$C. The maximum increase in performance of 300% over the smooth cylinder occurred when using the Turbotec tube and Freon 113 as working fluid.

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## ACKNOWLEDGEMENT

I want to thank Mr. Ken Mothersell, the laboratory supervisor, Mr. Tom Christian, the laboratory manager, and Mr. Ron Longueira, the laboratory technician for their technical advise and help during my thesis work. Special thanks go to Mr. John Moulton, the modelmaker who made up the condenser sections for this project. Without all their help it would have been impossible for me to finish this experimental thesis.

# I. INTRODUCTION

## A. THE ROTATING HEAT PIPE

The rotating heat pipe is a system for heat transfer; its application is heat removal from rotating machines. The heat pipe consists of an evaporator section, a condenser section and a working fluid to transfer the heat from one section to the other. (See Figure 1.)

During operation, the heat pipe is rotating about its longitudinal axis. The working fluid forms an annulus in the evaporator section. If heat is added to the evaporator section the working fluid evaporates and is driven, due to the pressure difference, into the condenser section, which is cooled from the outside. The vapor condenses and is driven back to the evaporator section by the hydrostatic pressure gradient in the condensate. The return of the condensate can be enhanced by using either a slight taper of the condenser wall or by using internal spiral fins.

## B. BACKGROUND

The rotating heat pipe used at the Naval Postgraduate School was initially designed and constructed in 1970. Until now several modifications were implemented. During different research programs a standard set of rotational speeds was used: 700, 1400 and 2800 RPM. Earlier research programs were involved in testing steel and copper cylinders, steel and copper truncated cones and a Noranda internally

finned tube. The copper truncated cone showed excellent performance but due to the very high cost of manufacture, this tube was replaced by the more economical internally finned one. Although different working fluids such as distilled water, ethanol and freon 113 were tried, the spiral Noranda tube was only tested with water.

## C. THESIS OBJECTIVES

The thesis objective was to experiment with new condenser surfaces and working fluids to find their influence upon heat transfer performance and to confirm the data collected by Wagenseil[1].

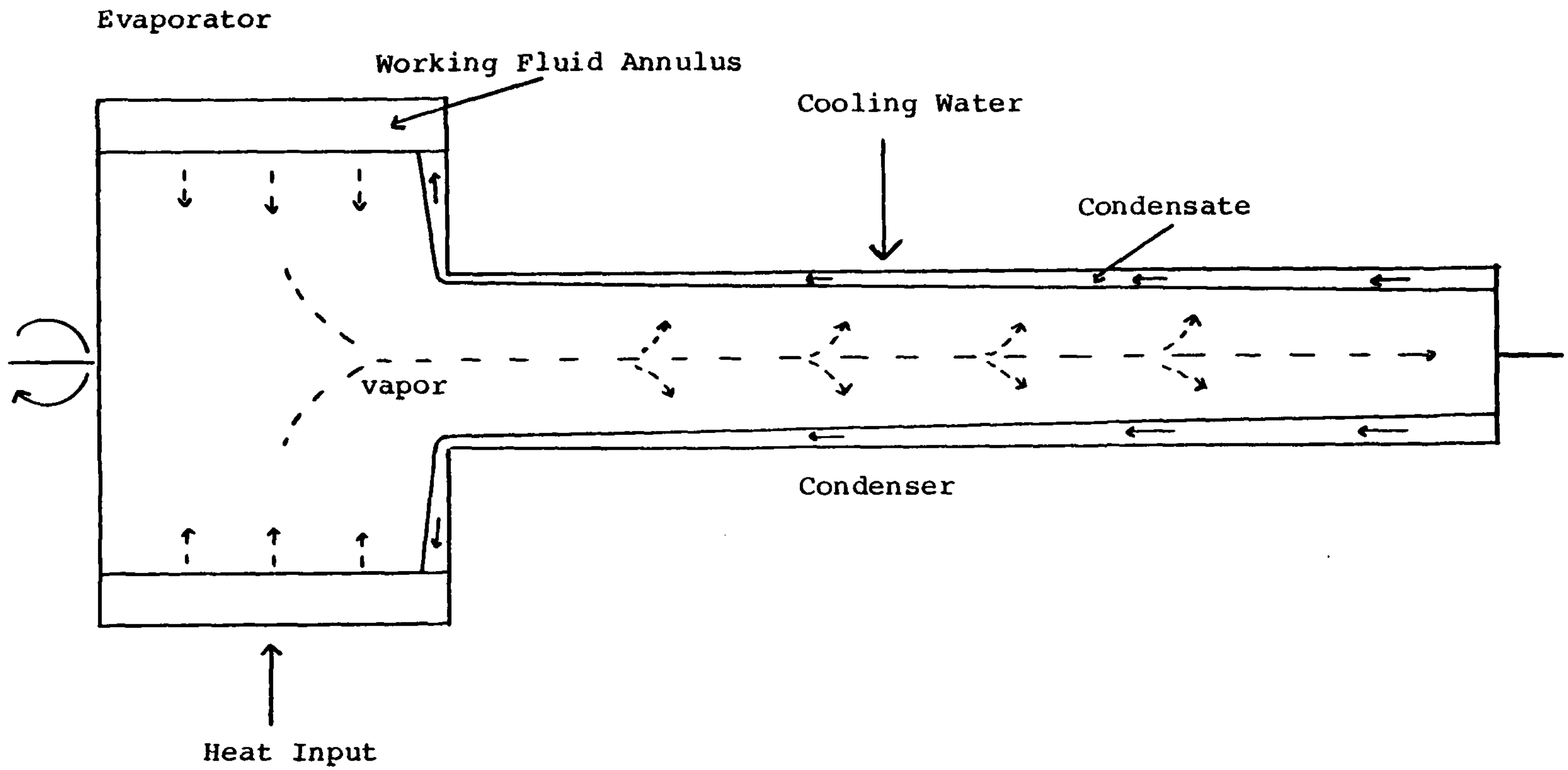



Figure 1. Working Principle of a Rotating Heat Pipe

## II. EXPERIMENTAL EQUIPMENT

### A. DESCRIPTION OF THE COMPONENTS

In the following paragraphs each component will be described (See Figure 2). The set-up is equivalent to that used by Wagenseil [1].

1. Evaporator

The cylindrical evaporator section is made out of copper and has O-ring seals on both ends.

2. Heater

A resistance heating element is wrapped around the evaporator section and then insulated. The ends of the heating element are connected to collectors to assure electrical power input via brushes.

3. Power Supply

A solid state phase amplifier power controller is used. The overall range of power was from 0.2 to 7.5 KW.

4. View Pieces

One side of the evaporator section is sealed with two round pyrex glass plates for visual observation of the vapor and condensate production.

5. Condenser Tube

On the other side of the evaporator an adapter is installed and inserted into a ball bearing, which is mounted on a steel plate holding the whole experimental set-up. The adapter piece has an O-ring seal, allowing interchangeable condenser tubes to be placed into the set-up. The condenser

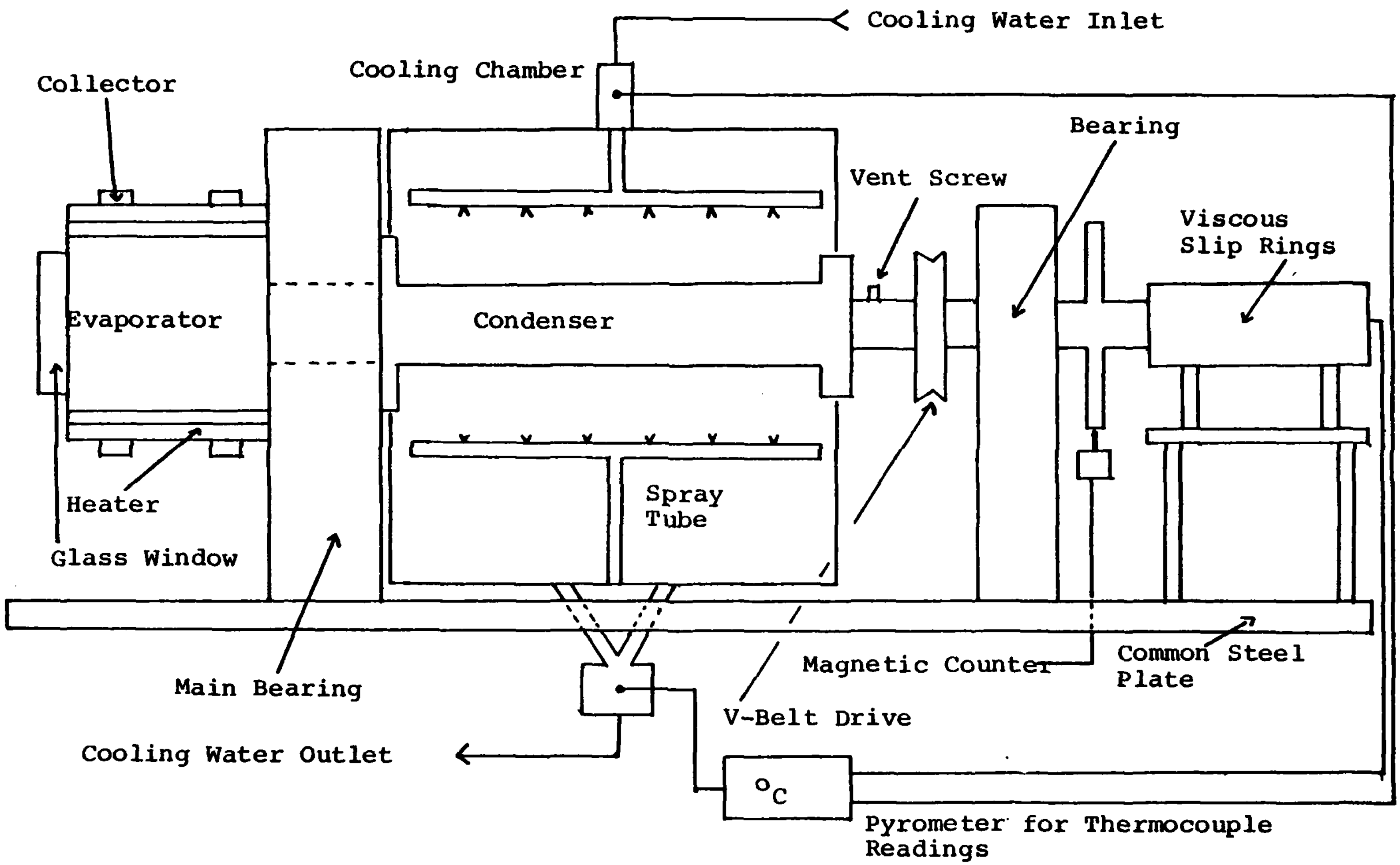



Figure 2. Experimental Set-Up

tube has a seal flange on each side, which is silver soldered into the copper tube for fastening purposes. The condenser tube is surrounded by an insulated cooling chamber.

6. Cooling System

Inside the cooling chamber there are 4 spray-arms, $90^{o}$ apart, to cool the entire length of the condenser. The water flow rate is adjusted to 50% on a rotameter. (See Figure 3).

7. Heat Pipe Drive System

In the other side of the condenser tube a driving shaft is inserted and also sealed by an O-ring. A V-belt is run over the drive shaft to a variable speed motor. The drive shaft is hollow up to a point where a hole is drilled into it. This hole is sealed by a vent screw. The thermocouples of the evaporator and condenser section are connected to a viscous slip ring system. Between the slip rings and the V-belt of the driving system a maintenance free bearing is located and mounted to the common steel plate.

8. Lubricating System

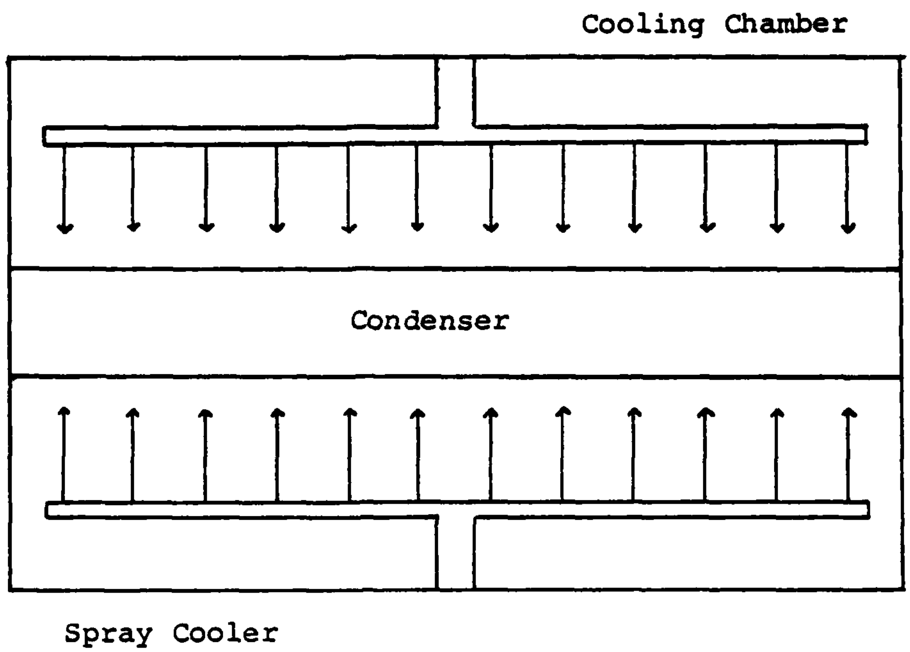
The main bearing at the evaporator section is lubricated with a drop oiler (one drop every 10 seconds).

9. Speed Control

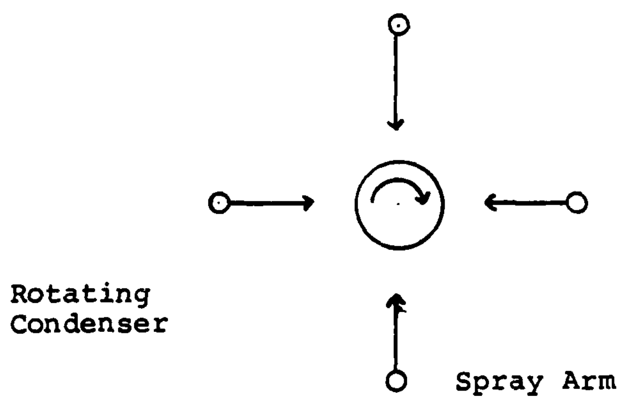
The drive shaft RPM is monitored by a magnetic tachometer which is digitally displayed.

10. Thermocouples

Two thermocouples are used to monitor the vapor temperature. Each is inserted into a steel tube which




(a) Side View




(b) Front View

Figure 3. Cooling Water System.

reaches five centimeters into the evaporator through drilled holes in the adapter piece. Nine thermocouples are located on the condenser tube wall to read the temperature distribution along its length. (See Figure 4). Two thermocouples are placed in the cooling water system. One monitors the water inlet temperature, the other the cooling water outlet temperature out of the mixer.

The evaporator and condenser themocouples are teflon insulated copper constantan wires; the water inlet and outlet thermocouples are made out of copper and constantan with plastic insulation. All temperature readings were taken on a digital pyrometer. The display is given in degrees Celsius. (accuracy $\pm$ 0.1 $^{\circ}$C)

11. Test Condensers

All condensers have the same length of 25.7 cm and are made out of copper.

a. Smooth Cylinder

Outer diameter: 29.0 mm

Wall thickness: 2.0 mm (See Figure 5)

b. Spiral Noranda Tube

Outer diameter: 26.5 mm

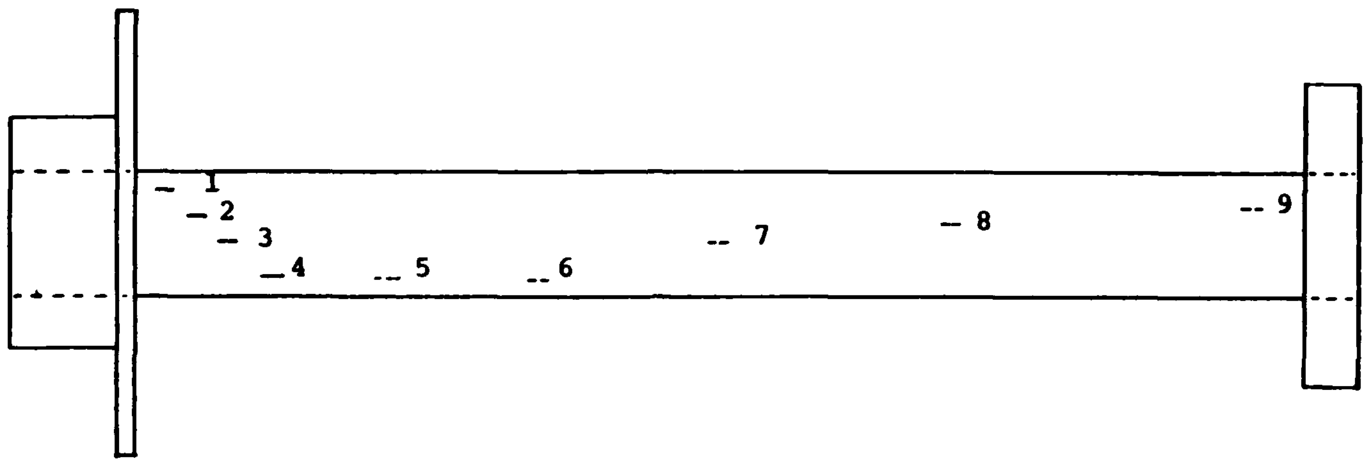
Wall thickness: 1.0 mm

Number of fins: 16

Height of fins: 1.5 mm

Pitch of fins: 152.4 mm

The inside area ratio of this tube to the smooth cylinder is 1.67. (See Figure 6).




| Thermocouple Number | Distance from Evaporator Wall, cm |
|---|---|
| 1 | 0.5 |
| 2 | 0.9 |
| 3 | 1.5 |
| 4 | 2.5 |
| 5 | 5.1 |
| 6 | 7.6 |
| 7 | 12.7 |
| 8 | 17.8 |
| 9 | 21.6 |

Figure 4. Position of Thermocouples on Condenser Tube.

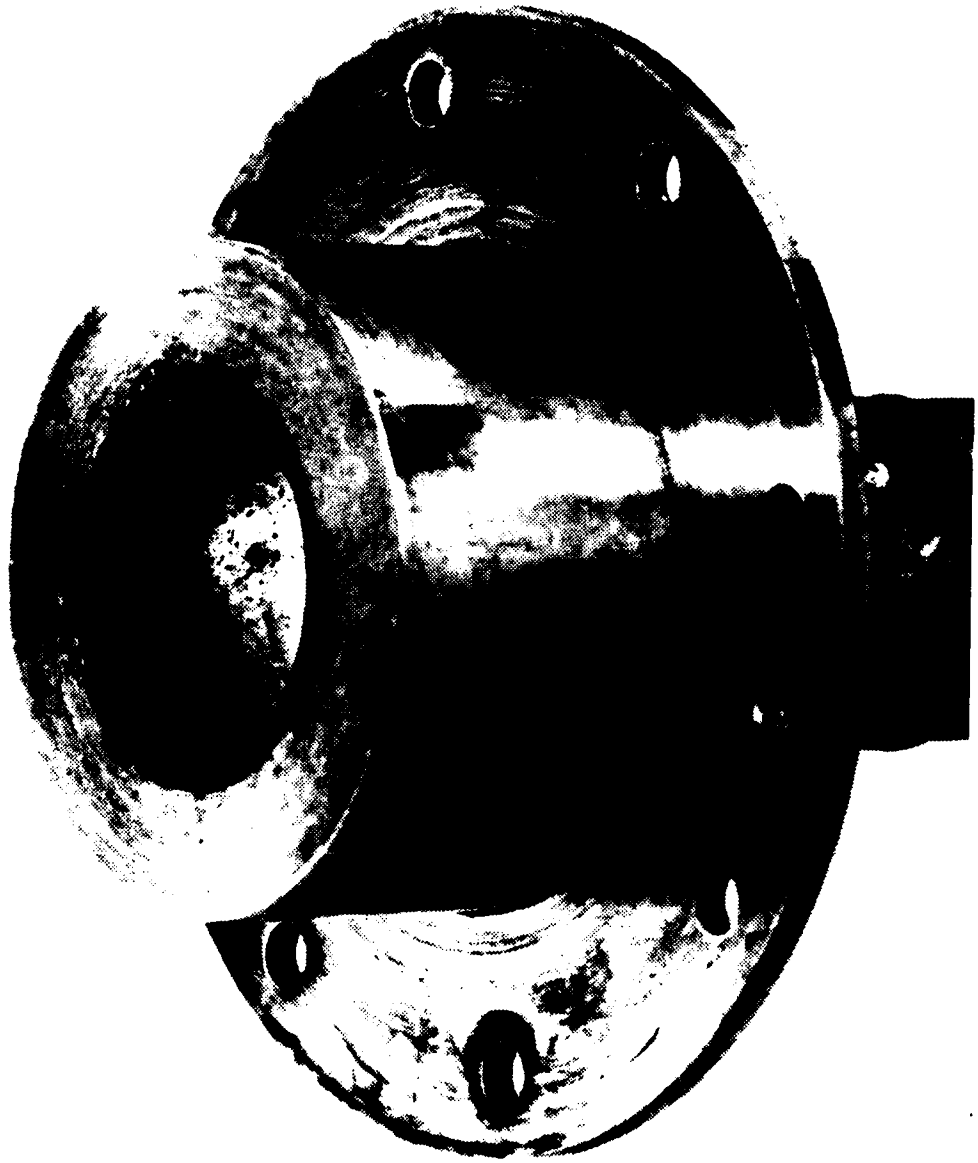


Figure 5. Photograph of the Smooth Cylinder

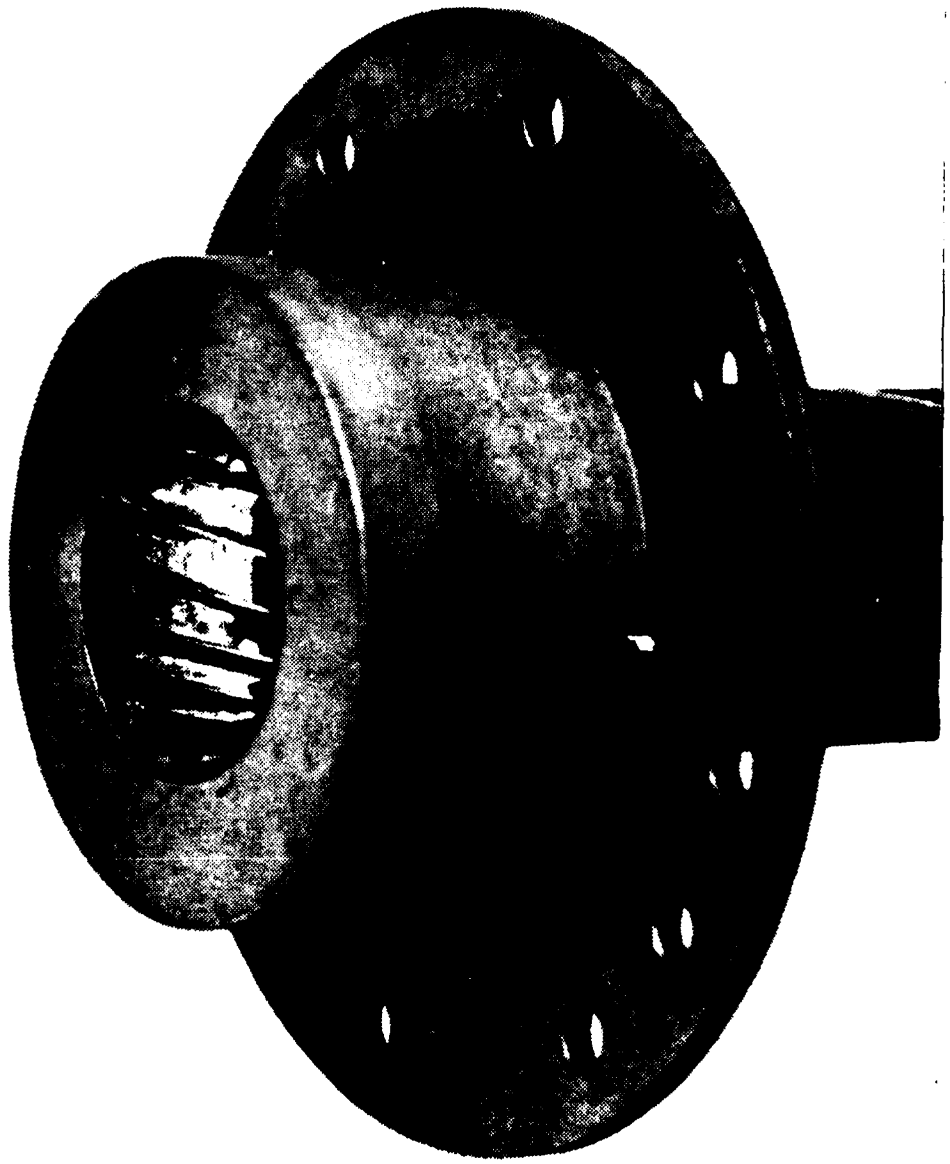


Figure 6. Photograph of the Spiral Noranda Tube

c. Hitachi Thermoexcel-C Tube

Outer diameter: 25.0 mm

Wall thickness: 1.8 mm (including enhanced surface). Treating this tube as a smooth cylinder, the area ratio is 0.9. The actual inside surface might have an area ratio of 2 to 3 (it is not possible to calculate it.). (See Figure 7).

d. Hitachi Thermofin Tube Type II

Outer diameter: 25.4 mm

Wall thickness: 0.8 mm

Number of fins: 50

Height of fins: 0.75 mm

Spacing of fins: 0.75 mm

The area ratio to the smooth cylinder is 1.91. (See Figure 8).

e. Turbotec Tube

Outer diameter: 27.5 mm

Wall thickness: 0.8 mm

Number of fins: 3

Height of fins: 5.0 mm

Pitch of fins: 97.0 mm

This tube is fabricated by twisting a smooth cylinder; the result is an inside and outside spiral tube. The area ratio to the smooth cylinder is 1.39. (See Figure 9).

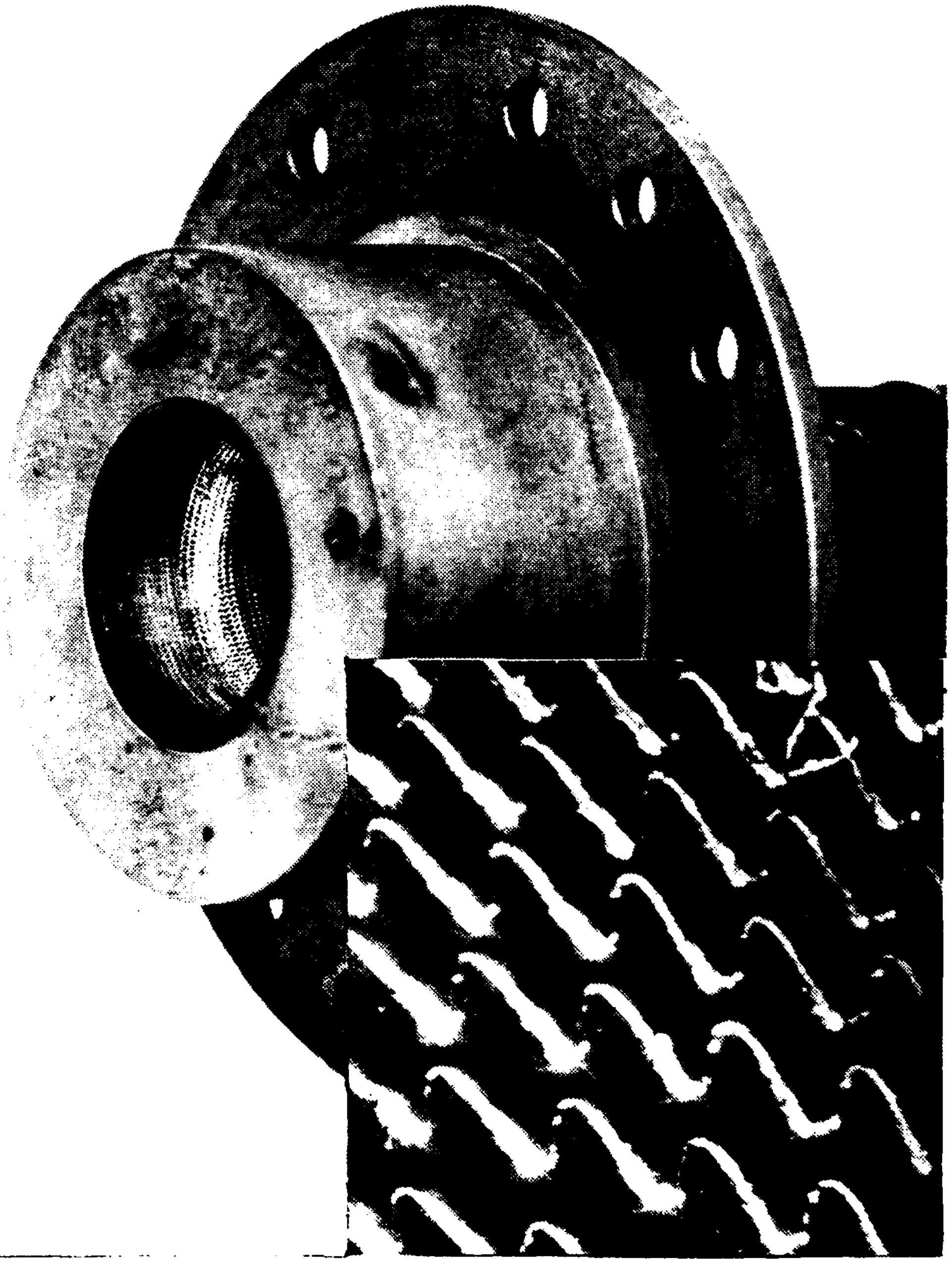


Figure 7. Photograph of the Hitachi Thermoexcel-C Tube

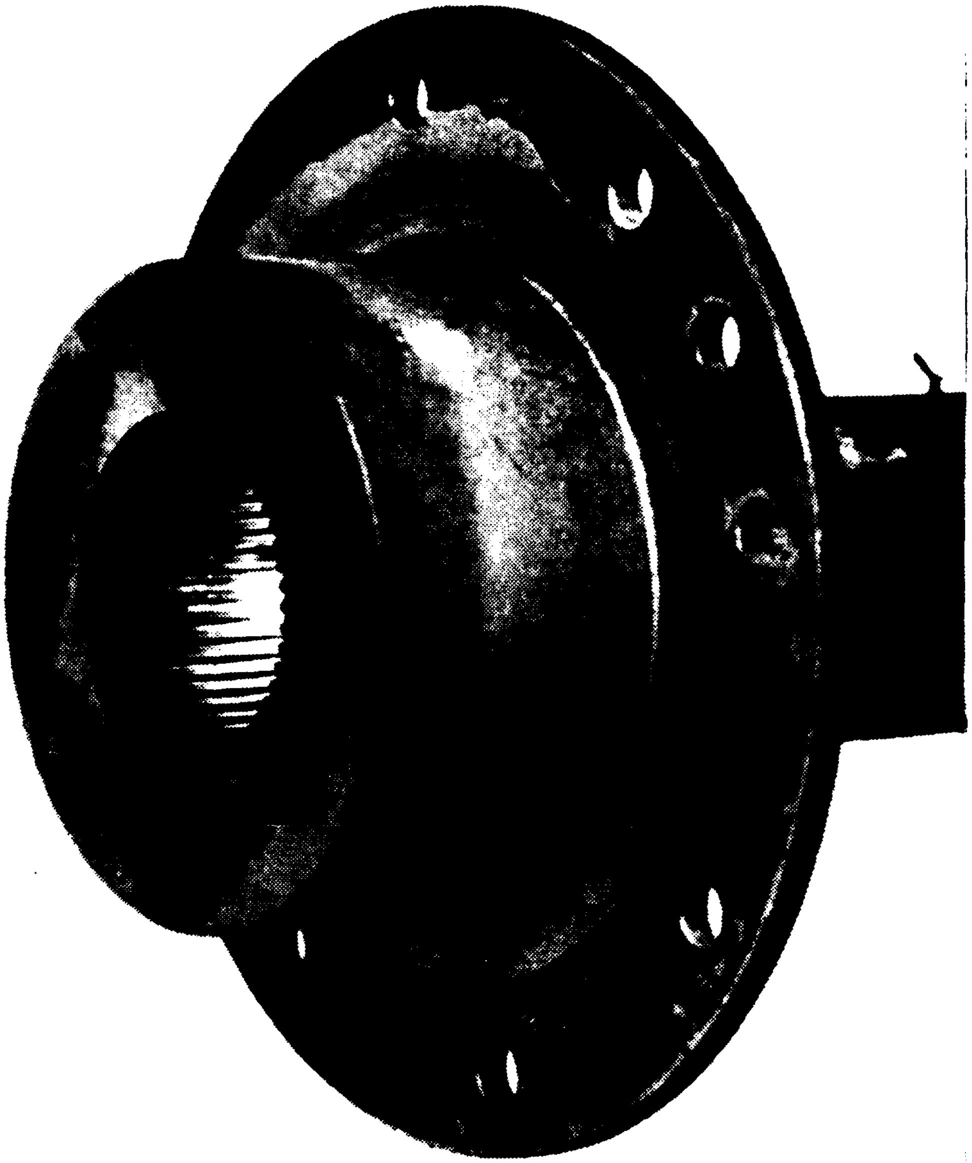


Figure 8. Photograph of the Hitachi Thermofin Tube Type II.

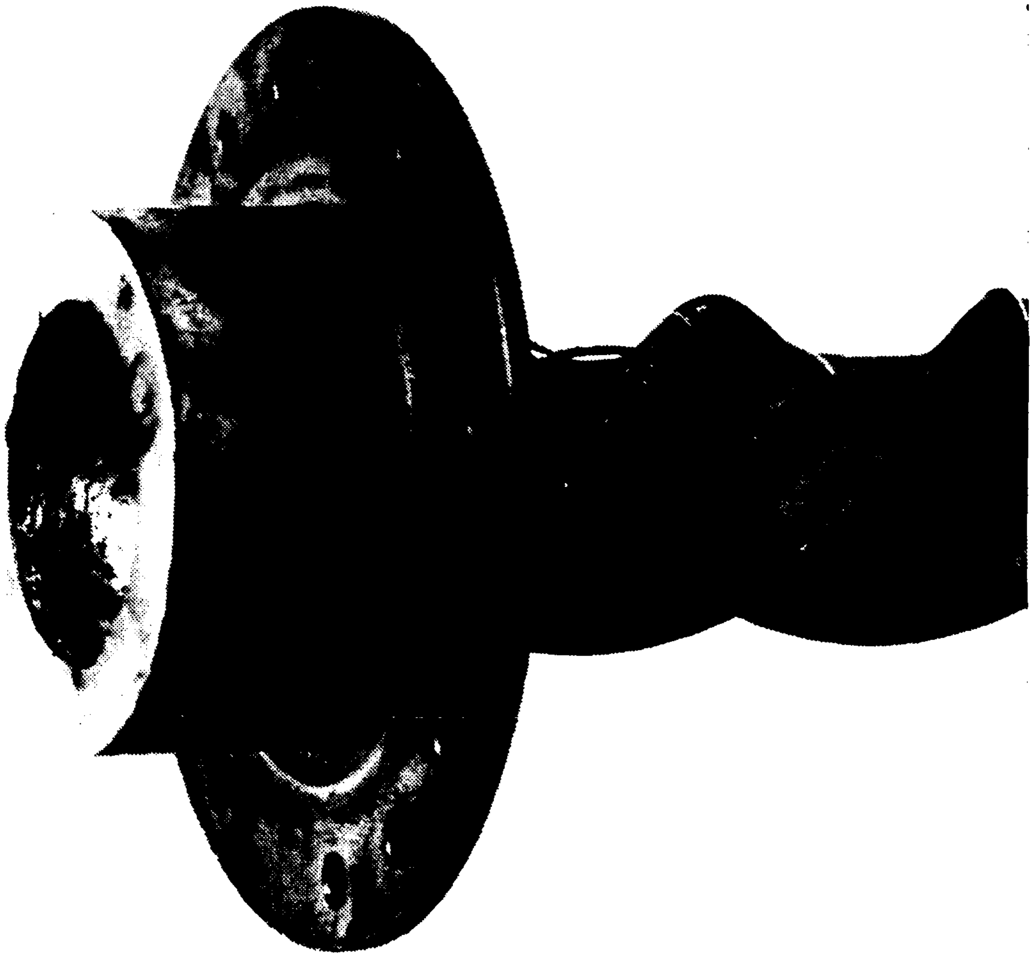


Figure 9. Photograph of the Turbotec Tube

# III. EXPERIMENTAL PROCEDURE

## A. PREPARATION OF CONDENSER WALL

Each condenser surface was treated chemically to assure filmwise condensation. The solution which was used consisted of equal parts of ethanol and a 50% solution of sodium hydroxide warmed to 80$^{\circ}$C. The condenser was scrubbed inside with the solution. The surface was well wetted after this procedure which is an indication for filmwise condensation. The condenser tube was then placed into the experimental set-up and checked for leakage by drawing a vacuum. For this, the vent screw was replaced by a hollow one to have access to the inside of the tube to draw a vacuum.

## B. FILL PROCEDURE

The common steel plate which mounts the whole experimental set-up was tilted about 60$^{\circ}$ with the evaporator section pointing upward. The two glass endpieces were removed and about 300 ml of working fluid were added. Then the glass pieces were replaced and tightened.

## C. VENTING PROCEDURE

The set-up was now tilted 45$^{\circ}$ downward so that the viscous slip rings were pointing upward. Heat was applied to the evaporator section until the vapor temperature reached a value 10$^{\circ}$C above boiling point of the working fluid at atmospheric conditions. The vent screw was removed and the

heat pipe was vented to the atmosphere until 100 ml of the working fluid was evaporated. This was necessary to drive all noncondensable gases out of the heat pipe. Then the vent screw was inserted and the power supply was shut off. The experimental set-up then was placed in a horizontal position to get it ready for running.

D. RUNNING PROCEDURE

Throughout this thesis the following running procedure was used:

a. Open cooling water supply and adjust flow rate.

b. Open main bearing cooling water valve.

c. Open needle valve of drop oiler.

d. Start variable speed motor. Run up to 1400 RPM to assure formation of liquid annulus in the evaporator.

e. Select desired RPM.

f. Set heater power to desired level. Wait for steady state.

g. Take thermocouple readings.

h. Repeat steps f and g until the saturation temperature reaches the boiling point of working fluid at atmospheric conditions.

i. Repeat steps a to i for all rotational speeds and working fluids.

Best repeatable results were achieved by reventing after each run.

## E. DATA REDUCTION

To evaluate the rate of heat transfer from the condenser, a heat balance was made using the basic equation

$$Q = \dot{m}c_p\Delta T$$

where

$\dot{m}$ = Cooling water flow rate

$c_p$ = Specific heat of cooling water

$\Delta T$ = Temperature difference between cooling water outlet and inlet

At a given RPM, to eliminate effects of frictional heat generated by the system, the value of the condenser heat transferred during each zero power run was subtracted from the heat transferred at any power setting.

# IV. PRESENTATION AND DISCUSSION OF RESULTS

## A. GENERAL COMMENTS

The term "performance" will be used and stands for the heat transfer rate achieved for a given condenser. Improvement in "performance" implies an increase in the heat transfer rate. Rotational speed is designated as RPM.

The results are presented in the following form. One condenser type will be compared in its performance using three different working fluids. The performance will also be compared at one saturation temperature minus cooling water inlet temperature ($T_s - T_{ci}$) of 40$^{o}$C and an improvement over the smooth cylinder will be given in percent increase or decrease in performance.

## B. RESULTS WITH THE SMOOTH CYLINDER

The smooth cylinder was chosen to provide a basis for comparison with the other types of tubes. Figures 10, 11 and 12 show the performance with distilled water, ethanol and Freon 113. In general, the performance increases with increasing RPM. Leppert and Nimmo[2,3] have shown that the heat transfer coefficient for condensation in a rotating cylinder increases with RPM specifically as:

$$Nu_m \propto RPM^{2/5}$$

where $Nu_m$ is the mean Nusselt number .

The improvement is due to the increased centrifugal force on the condensate, which flattens the condensate film and

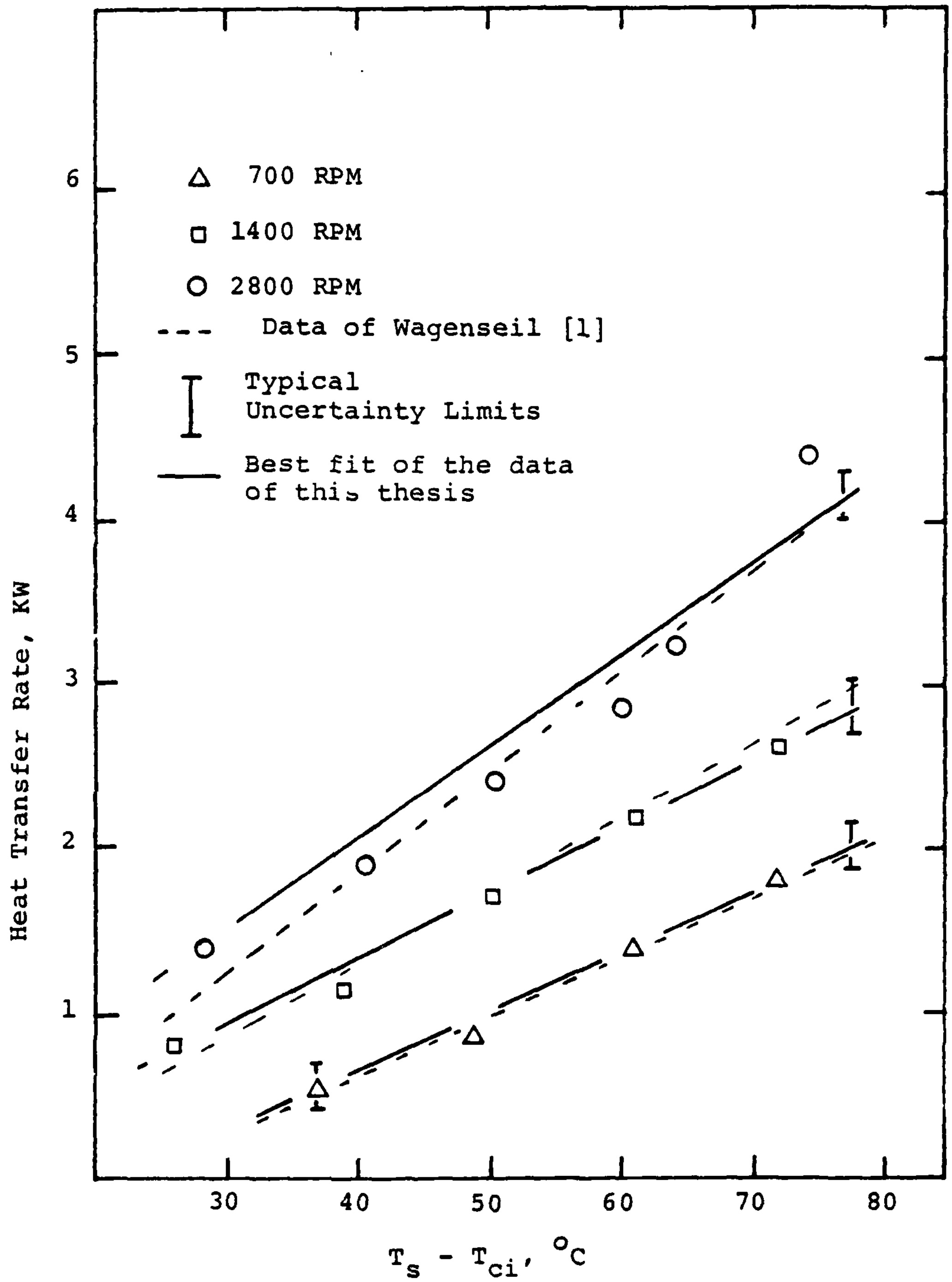



Figure 10. Heat Transfer Rate Versus Saturation Temperature Minus Water Inlet Temperature for Smooth Wall Cylinder and Distilled Water.

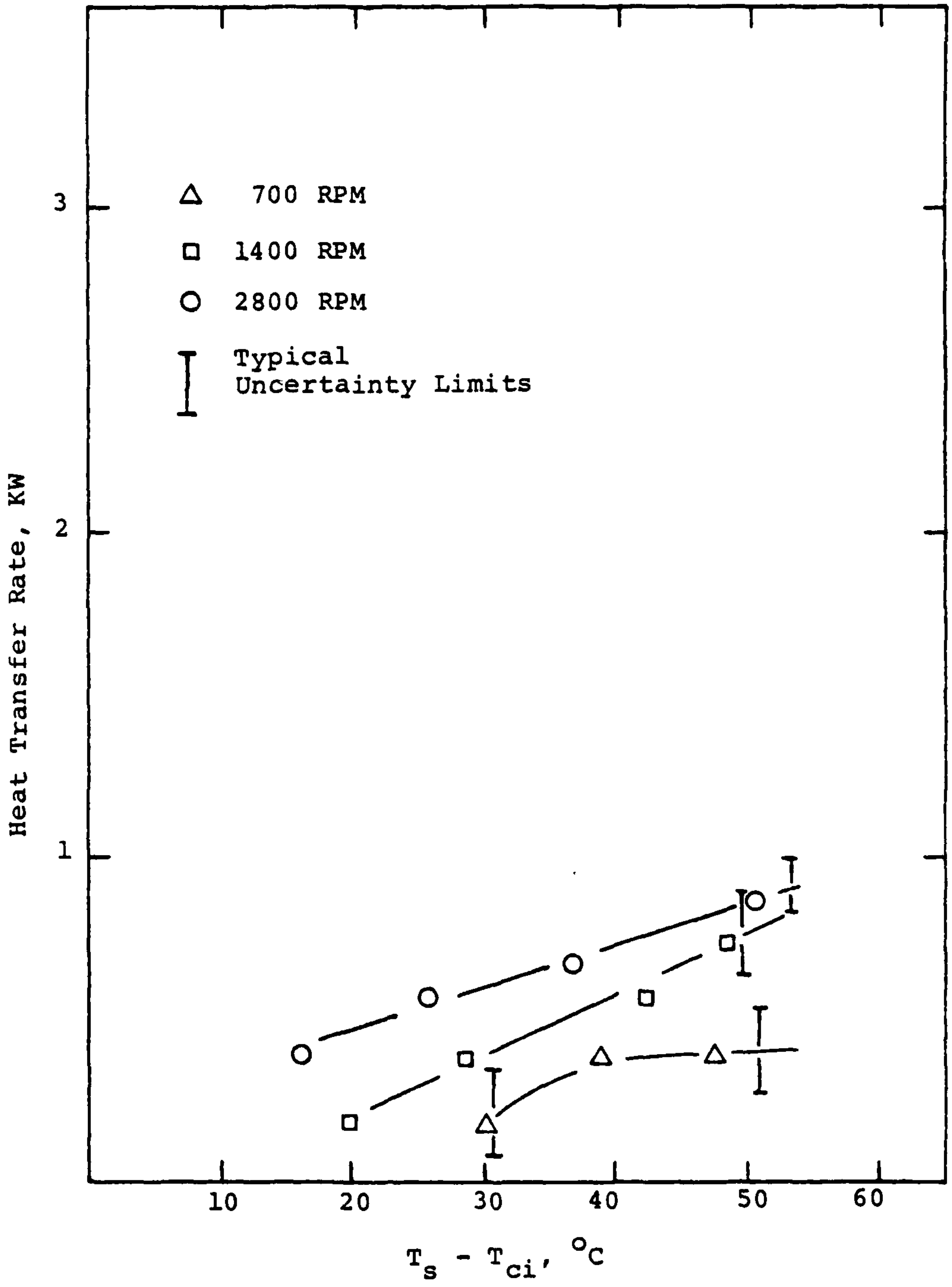



Figure 11. Heat Transfer Rate Versus Saturation Temperature Minus Water Inlet Temperature for Smooth Wall Cylinder and Ethanol.

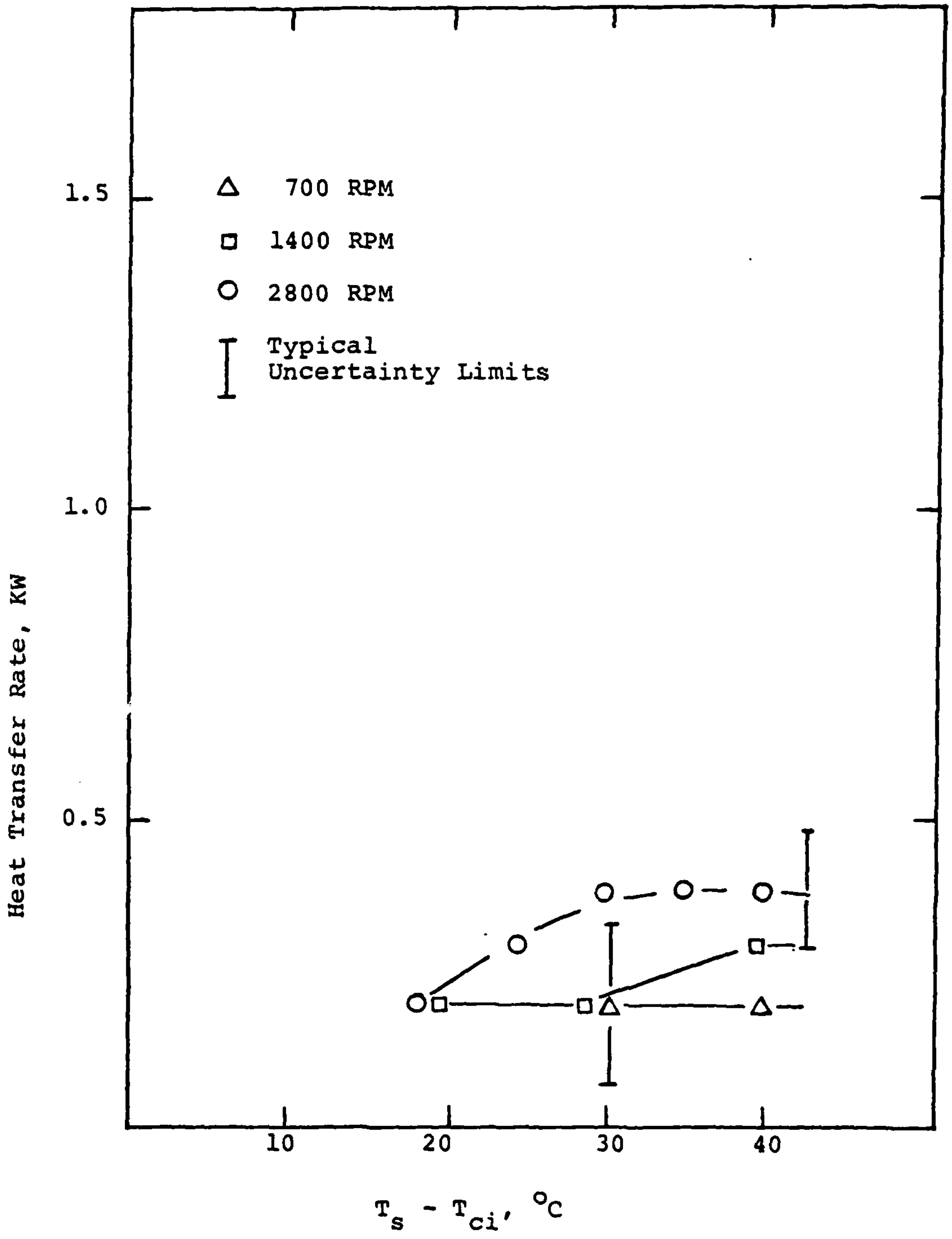



Figure 12. Heat Transfer Rate Versus Saturation Temperature Minus Water Inlet Temperature For Smooth Wall Cylinder and Freon 113.

thereby reduces the overall thermal resistance in the condenser. The performance of the smooth cylinder with distilled water as working fluid is superior to ethanol and Freon 113 as working fluid, due to the very good heat transfer performance of distilled water. Ethanol has the second best heat transfer performance. At ($T_s$ - $T_{ci}$) of 40°C, ethanol has 70% less, Freon 113 80% less heat transfer performance than distilled water. The results with the smooth cylinder and distilled water are in close agreement with those achieved by Wagenseil [1].

## C. RESULTS WITH THE SPIRAL NORANDA TUBE

Figures 13, 14 and 15 show the results with the spiral Noranda tube. The performance increases with increasing RPM for all three working fluids. The performance curves lie above the smooth cylinder curves. The results with this tube and distilled water are also in good agreement with those achieved by Wagenseil [1]. At ($T_s$ - $T_{ci}$) of 40°C, the increase for the distilled water is 100%, for ethanol 200% for all three RPM, and for Freon 113 there is an increase of 175% at 700 RPM, 250% at 1400 RPM and 290% at 2800 RPM. This drastic increase in performance is not possible only by the increased inside surface area (area ratio 1.64). The rest of the improvement is due to the pumping action of the internal spiral fins, which are driving the working fluid back into the evaporator. The higher increase in performance for ethanol and Freon 113 can be explained by looking at the different enthalpies of

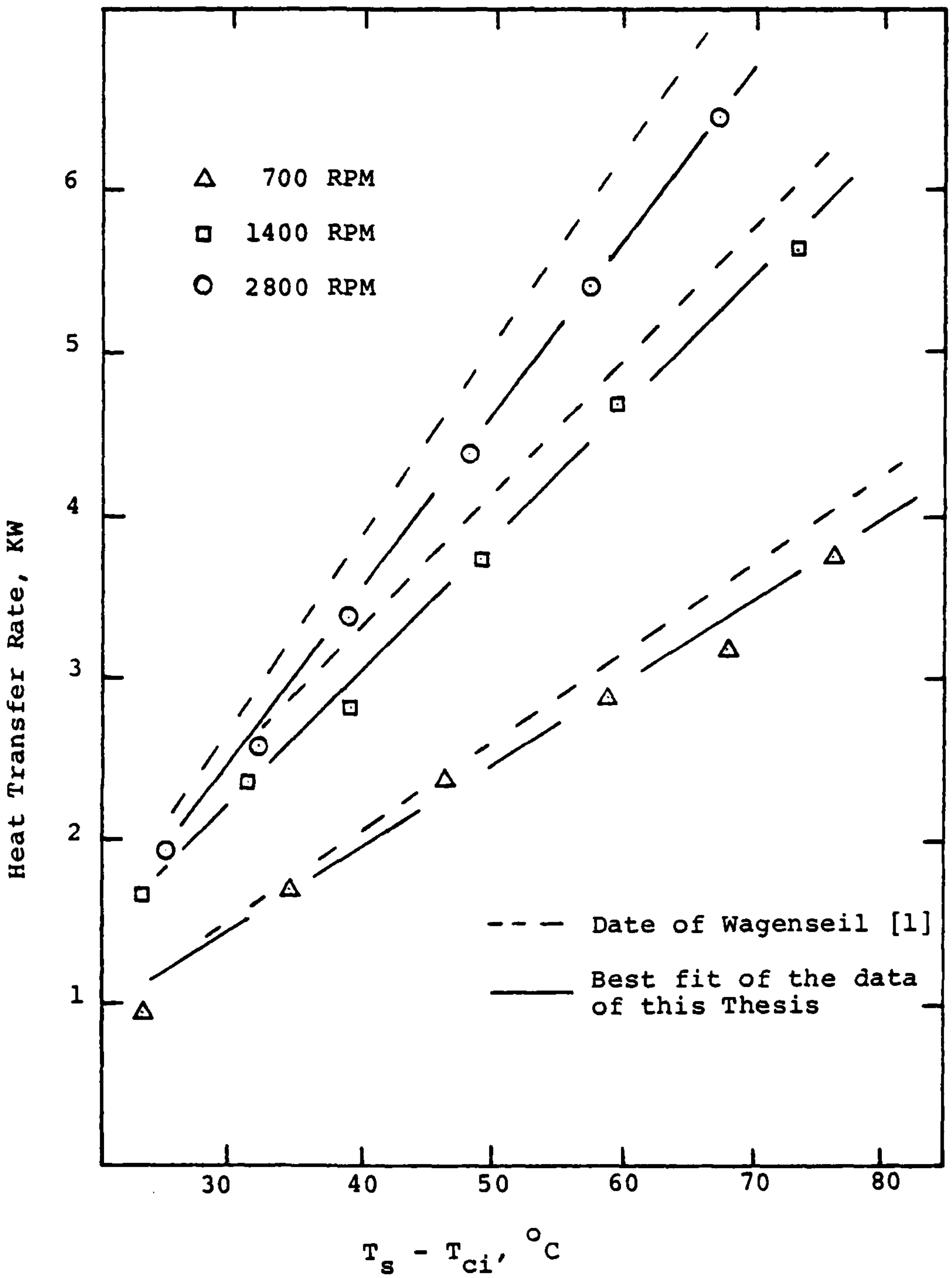



Figure 13. Heat Transfer Rate Versus Saturation Temperature Minus Water Inlet Temperature For Spiral Noranda Tube and Distilled water.

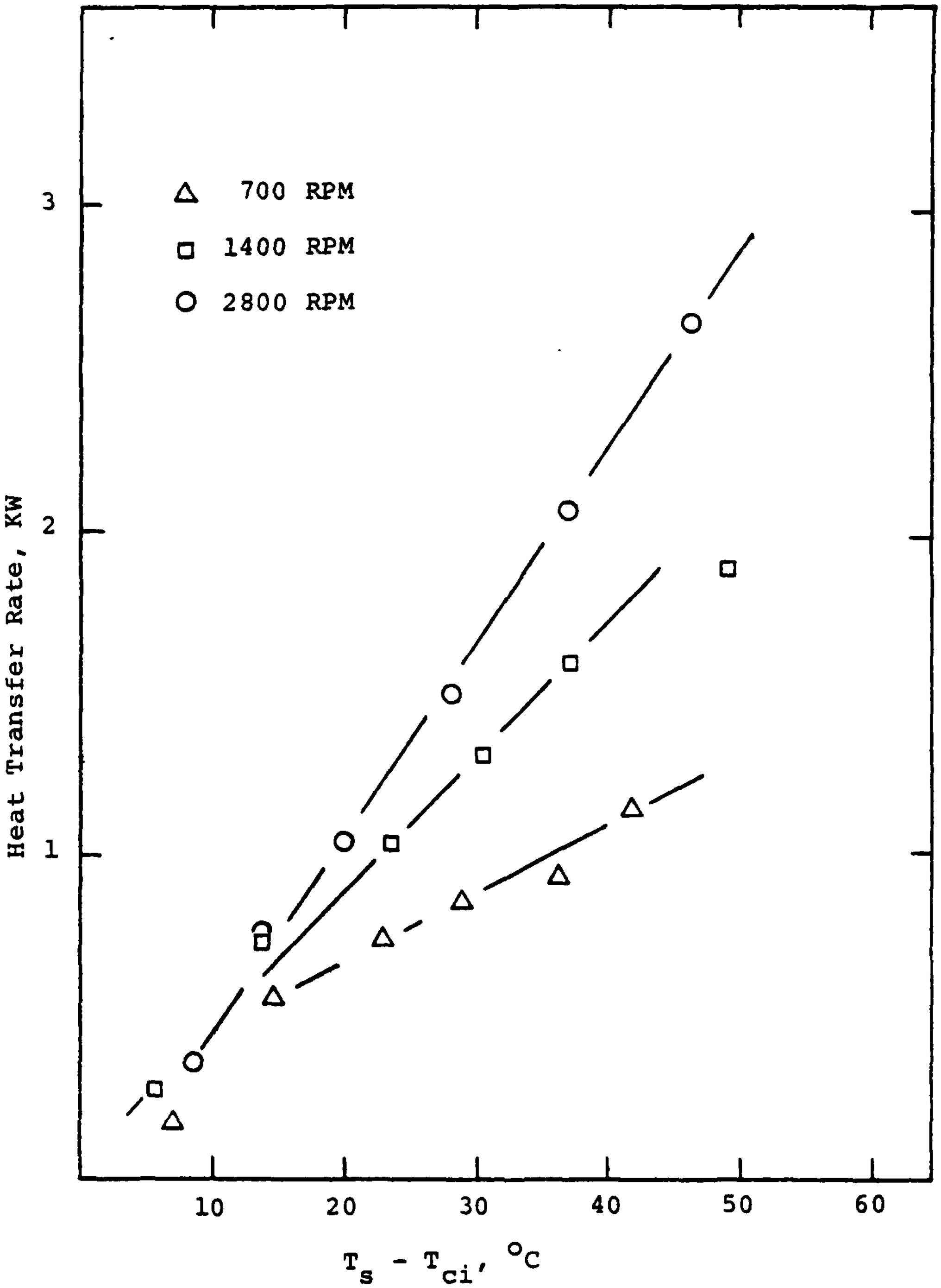



Figure 14. Heat Transfer Rate Versus Saturation Temperature Minus Water Inlet Temperature for Spiral Noranda Tube and Ethanol.

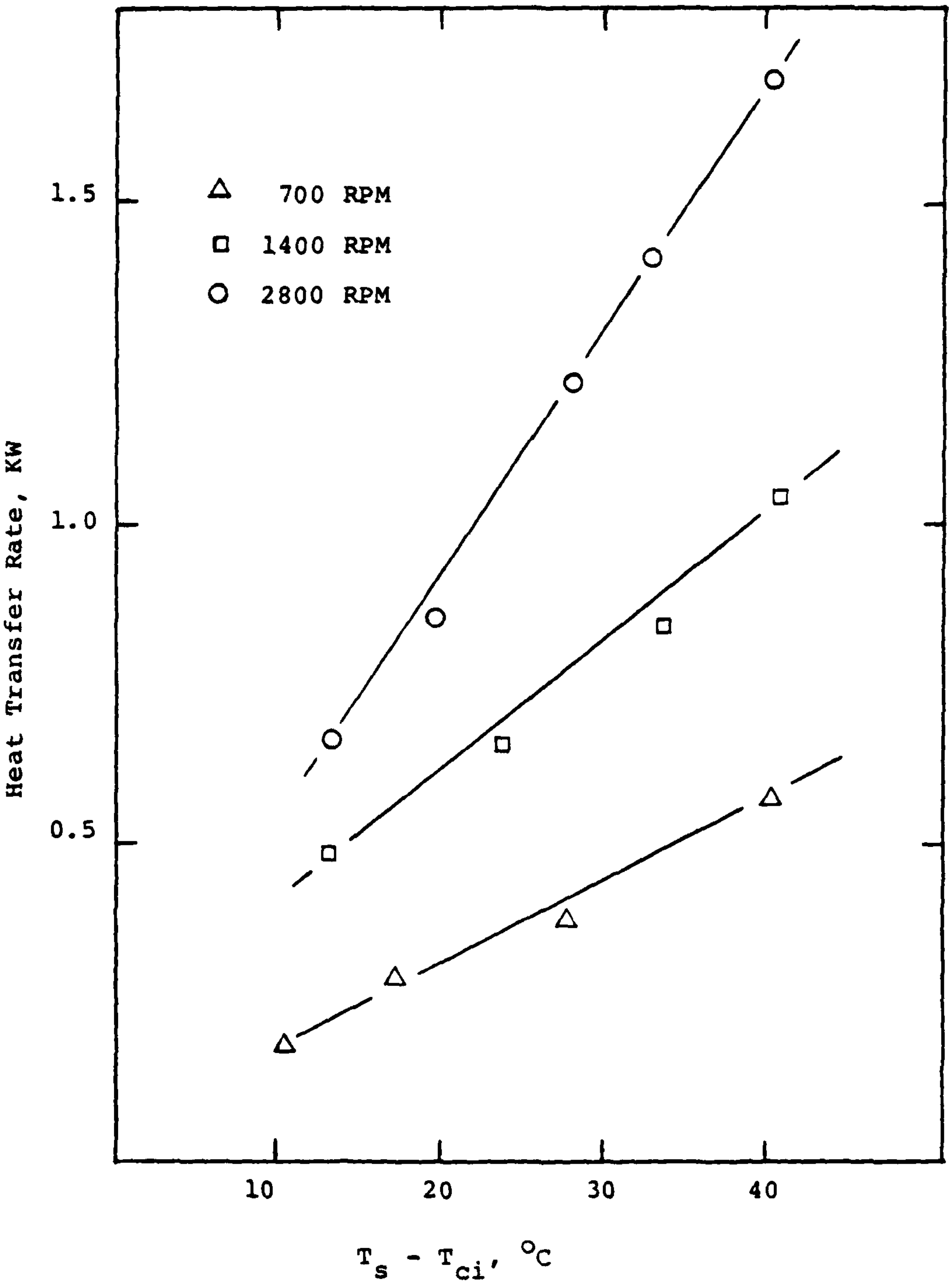



Figure 15. Heat Transfer Rate Versus Saturation Temperature Minus Water Inlet Temperature for Spiral Noranda Tube and Freon 113.

vaporization. At ($T_s$ - $T_{ci}$) of 40°C the heat input rate is in the same range for all three working fluids. But due to the fact that the enthalpy of vaporization of distilled water is about three times higher than for ethanol and about 16 times higher than for Freon 113 the mass rate of vapor produced in the case of Freon 113 is about 16 times and of ethanol about 3 times higher compared with distilled water. This different amount of vapor to be condensed leads to the observed percent increases in performance. Looking at this from another point of view, the heat transfer rate is proportional to the overall heat transfer coefficient. The reciprocal value of this coefficient is proportional to the sum of the reciprocal values of the inside and outside heat transfer coefficient and the thermal resistance of the wall. In comparing the three working fluids, the only variable is the inside heat transfer coefficient, which is a function of the pumping action, the fluid thermal conductivity, and the mass rate of vapor produced. Due to the high value of this inside coefficient for distilled water the governing resistance in this case is the outside heat transfer coefficient. The inside heat transfer coefficient for Freon 113 is much less than for distilled water; therefore, the inside coefficient is here the governing one. That means that an enhanced inside surface area and the pumping action will have much more effect on the actual heat transferred in the case of Freon 113. Ethanol lies in between these two discussed working fluids.

## D. RESULTS WITH THE HITACHI THERMOEXCEL-C TUBE

Figures 16, 17 and 18 show the results with the Hitachi Thermoexcel-C tube. Again, there is an increase in performance with RPM for all three working fluids. The increase is less significant, however, for ethanol and freon. In comparison with the values achieved using the smooth cylinder, the distilled water shows nearly the same heat transfer performance, the ethanol and the freon have a decrease in performance of 30 to 50% over the range from 700 to 2800 RPM. The reason for the bad performance of this tube is probably because this enhanced inside surface (See Figure 7) does not allow the backflow of the condensate to the evaporator until the little cavities are filled up with it. That means that the thermal resistance increases as the cavities continue to be filled until the backflow of the condensate starts. From there on this tube acts like a smooth cylinder. Due to the good heat transfer performance of distilled water the layer of water filling the cavities has no significant influence in the total performance. This now "smooth" cylinder has an area ratio of 0.9, that means nearly the same area is available as the smooth cylinder and therefore the performance of both these tubes with distilled water as working fluid show nearly the same value. The heat transfer performance of ethanol and Freon 113 is much less than with the smooth tube since the layer of working fluid inside the cavities acts like an insulation which reduces the heat transfer performance of this tube compared to the smooth cylinder. The thermal conductivity of water is 3.5 times that of ethanol and 5.8 times that of Freon 113.

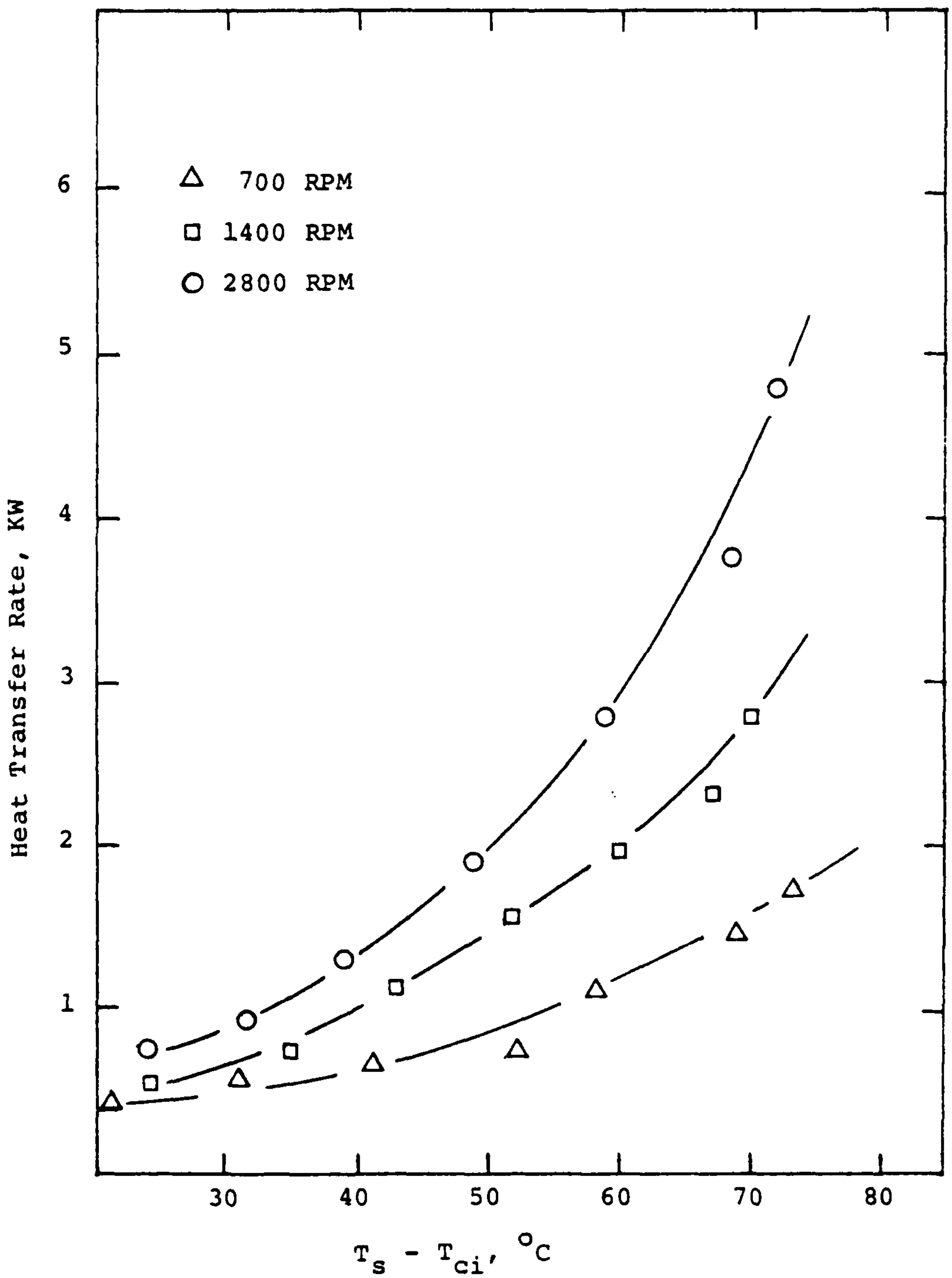



Figure 16. Heat Transfer Rate Versus Saturation Temperature Minus Water Inlet Temperature for Hitachi Thermoexcel-C Hitachi Tube and Distilled Water.

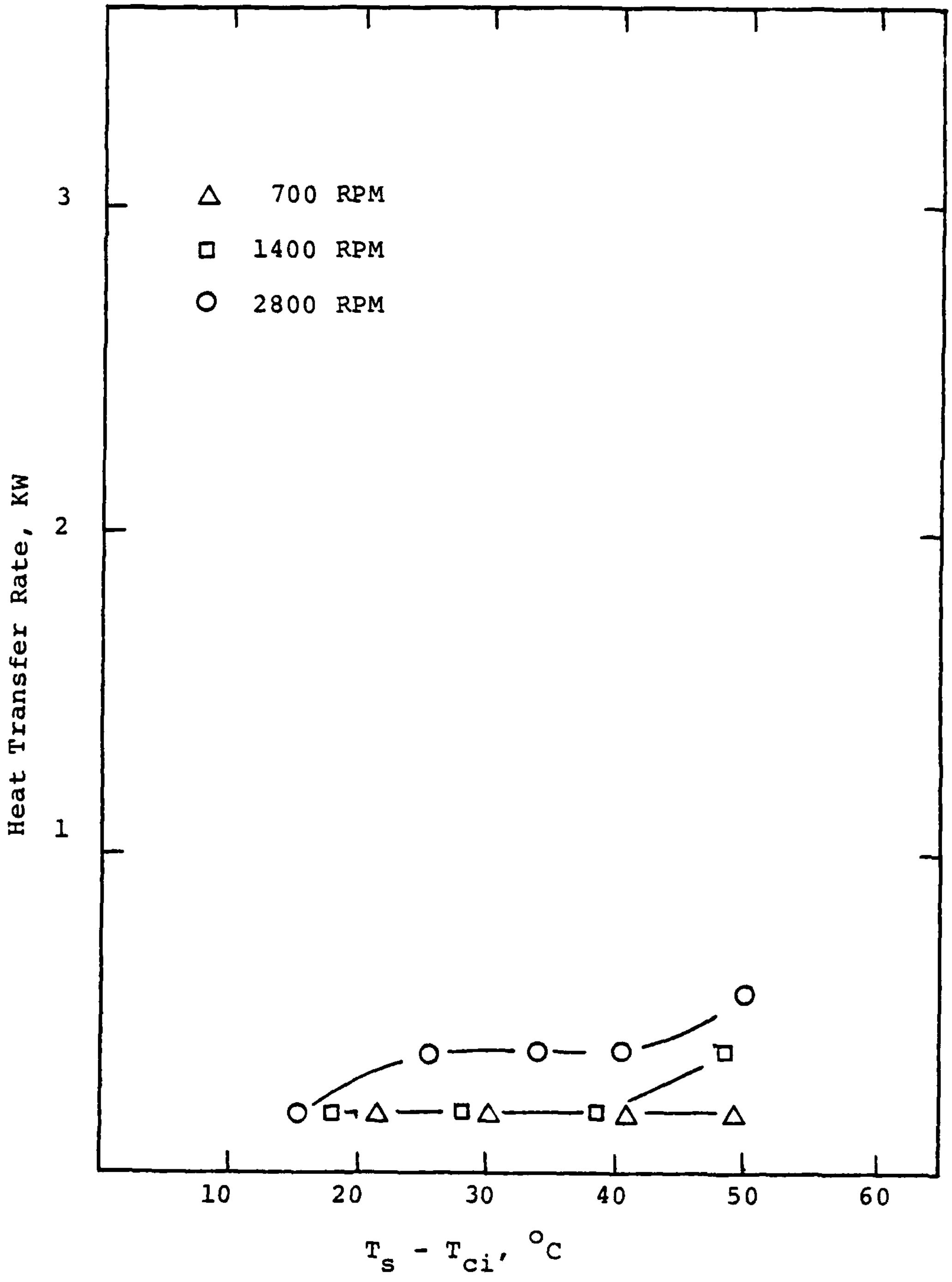



Figure 17. Heat Transfer Rate Versus Saturation Temperature Minus Water Inlet Temperature for Hitachi Thermoexcel-C Tube and Ethanol.

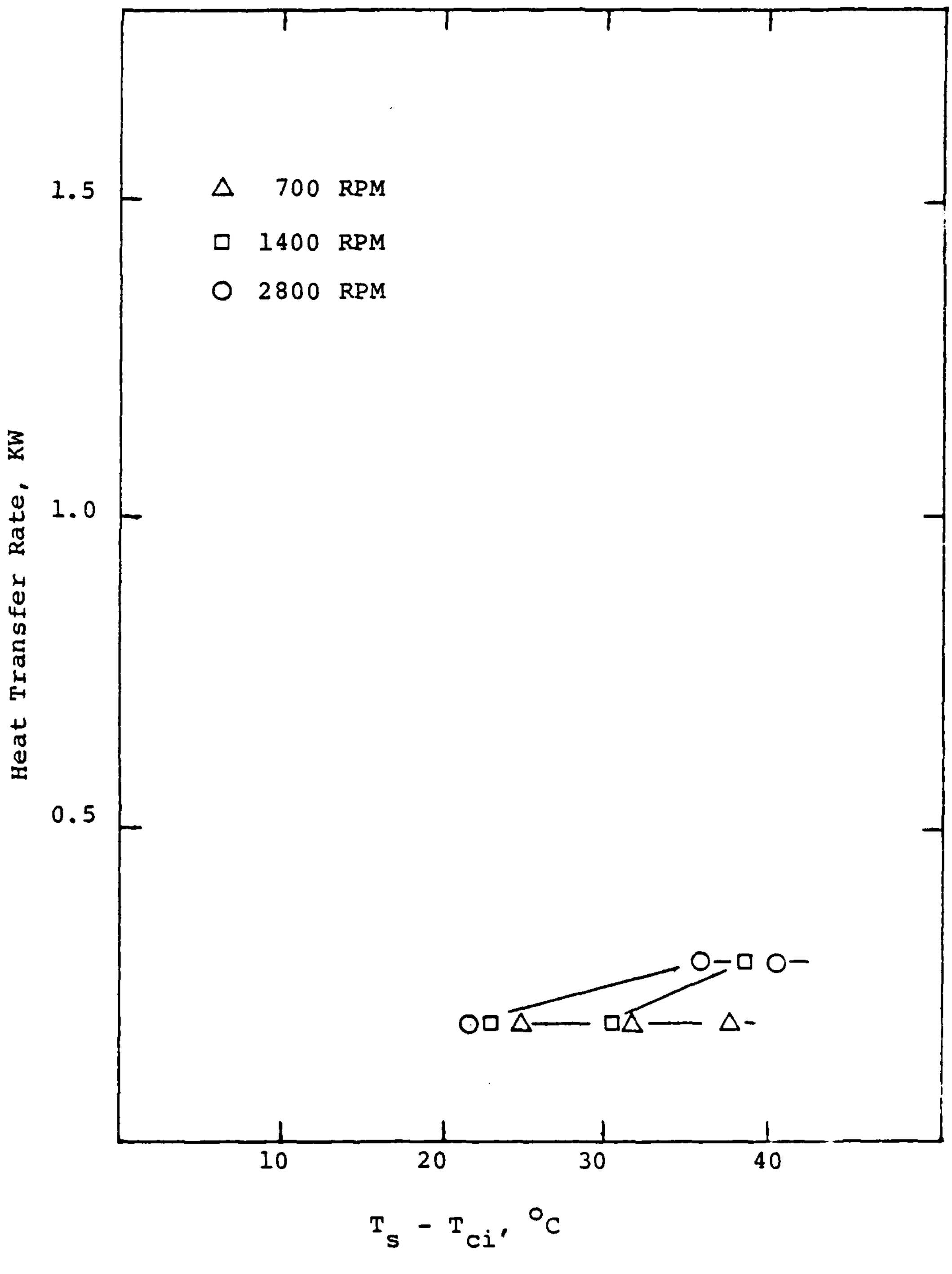



Figure 18. Heat Transfer Rate Versus Saturation Temperature Minus Water Inlet Temperature for Hitachi Thermoexcel-C Tube and Freon 113.

## E. RESULTS WITH THE HITACHI THERMOFIN TUBE TYPE II

Figures 19, 20 and 21 show the results with the Hitachi Thermofin Tube Type II. An increase in performance with RPM is given for all three working fluids. There is also an improvement in performance in comparison with the smooth cylinder. This improvement was expected due to the inside surface area ratio of 1.91. This area ratio would suggest an increase in performance of about 90%. The actual increase is 80% for distilled water, 60% for ethanol and only 40% for Freon 113. The less increase in performance has the following cause. The condensate is collected in the channels between the fins especially in the cases of ethanol and Freon 113, and forms a layer of fluid of not negligible thickness, which reduces the inside surface area for heat transfer. The thickness of the layer has different effects on the heat transfer due to the different thermal conductivities of the working fluids. The heat transfer is also reduced due to the increase of wall thickness in the area of the fins.

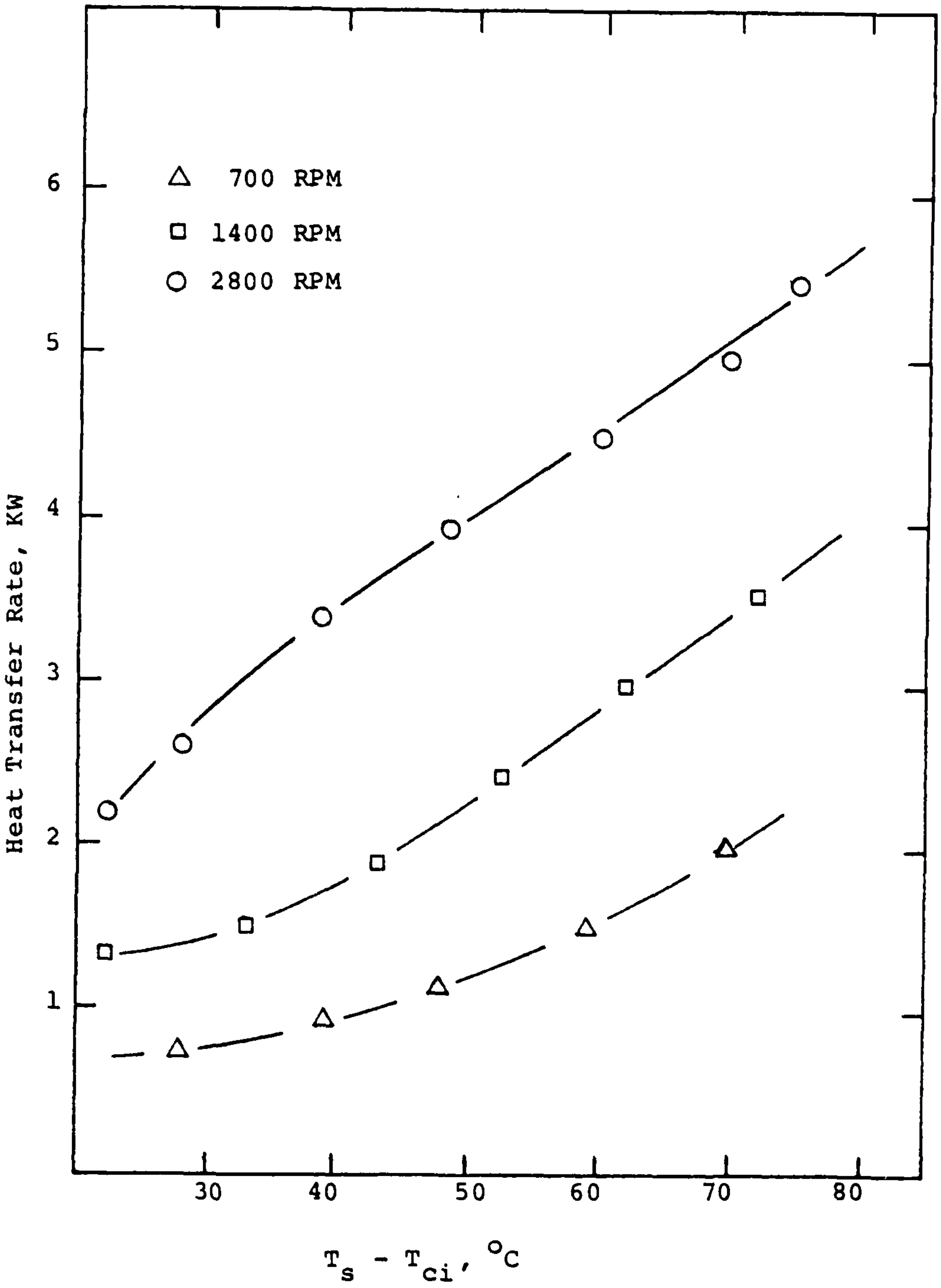



Figure 19. Heat Transfer Rate Versus Saturation Temperature Minus Water Inlet Temperature for Hitachi Thermofin Tube Type II and Distilled Water.

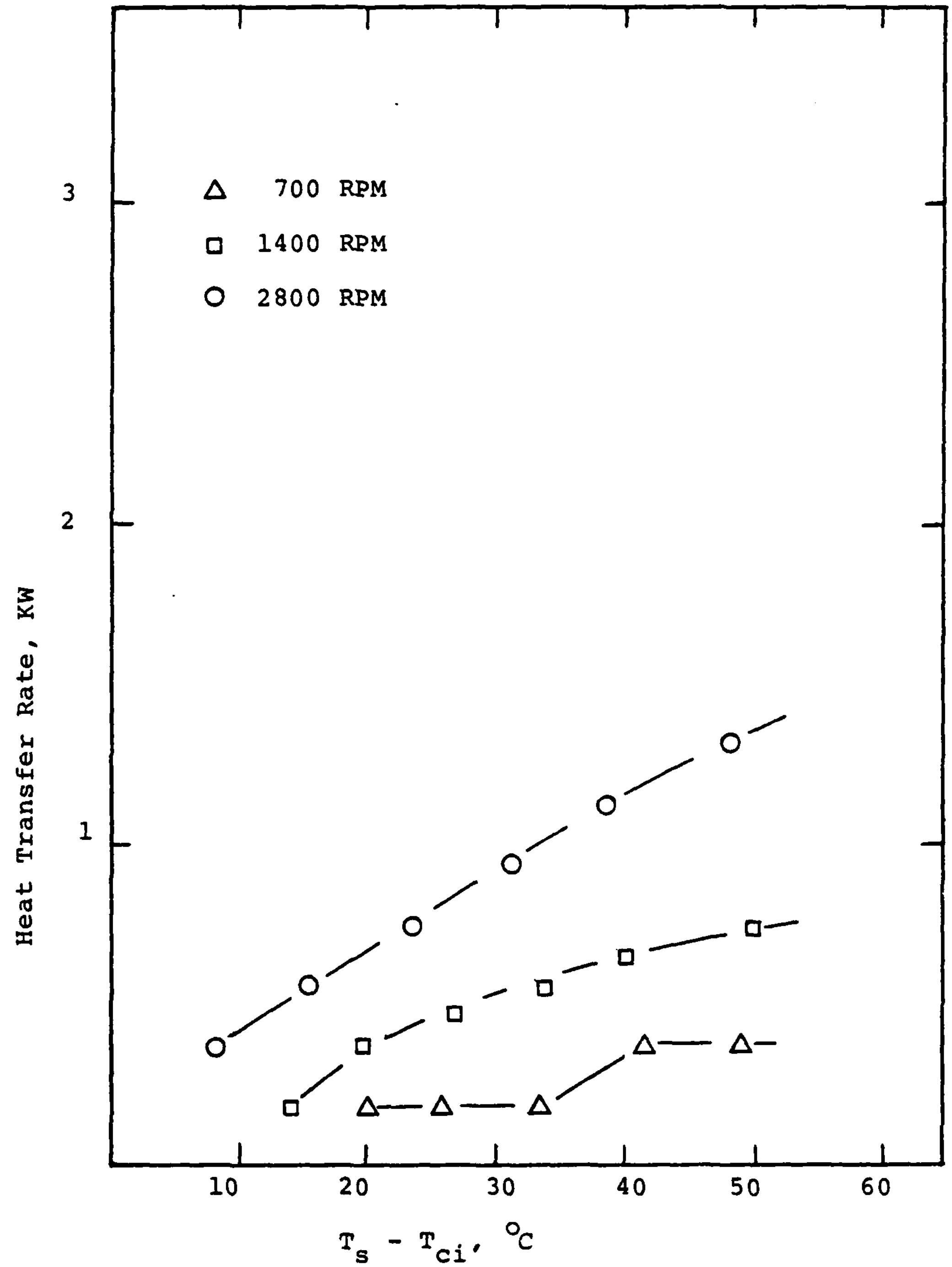



Figure 20. Heat Transfer Rate Versus Saturation Temperature Minute Water Inlet Temperature for Hitachi Thermofin Tube Type II and Ethanol.

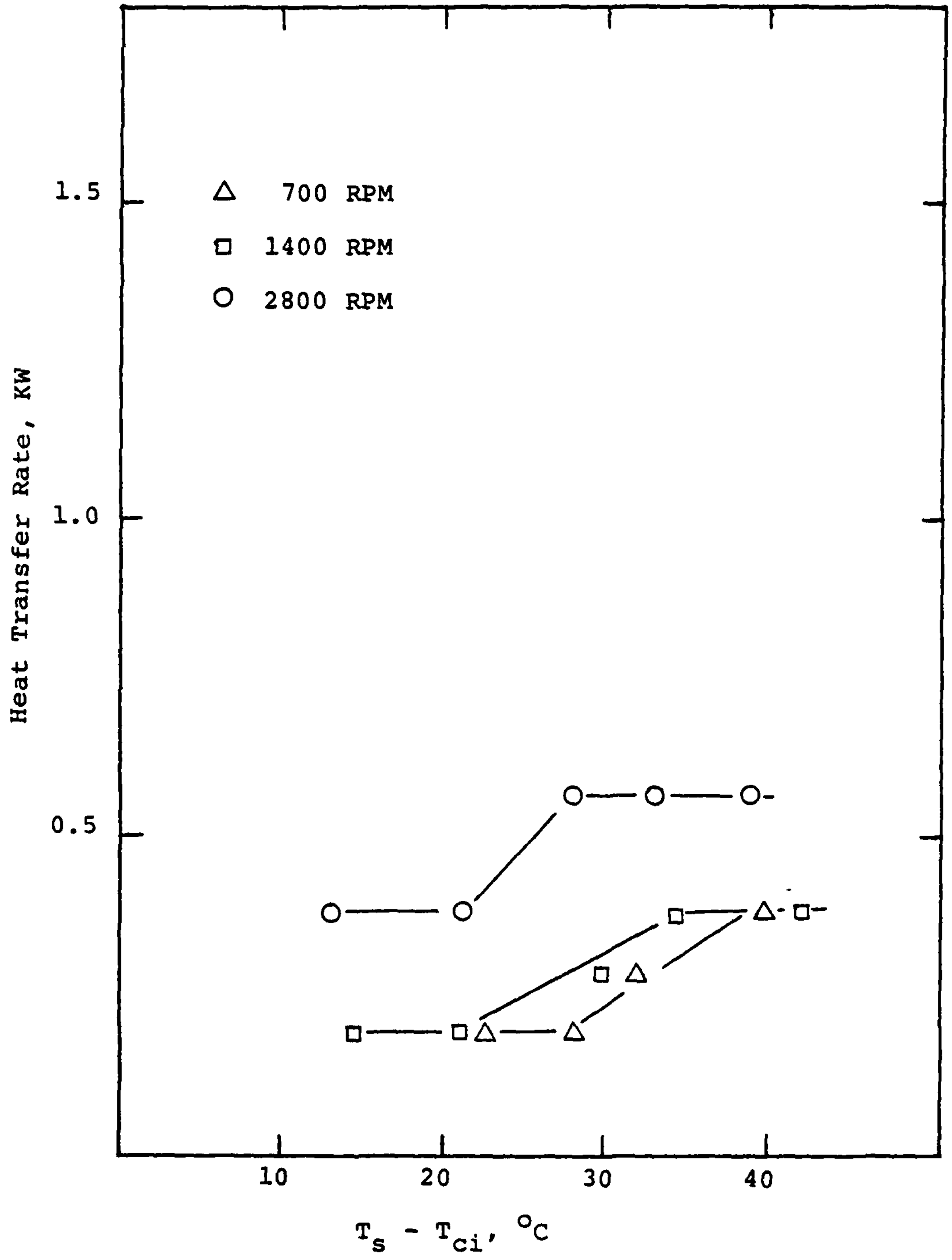



Figure 21. Heat Transfer Rate Versus Saturation Temperature Minus Water Inlet Temperature for Hitachi Termofin Tube Type II and Freon 113.

## F. RESULTS WITH THE TURBOTEC TUBE

Figures 22, 23 and 24 show the results with the Turbotec tube. A significant difference in the performance curves can be seen. In the case of distilled water an increase in heat transfer is obvious from 700 to 1400 RPM, but the 2800 RPM curve starts lower than the other two with a very low slope until about 40 to 45°C below the boiling point at atmospheric conditions. From there on the slope increases significantly exceeding the 700 and 1400 RPM slopes. In the case of ethanol the heat transfer increases from 700 to 1400 RPM, but the 2800 RPM curve starts at the same value as the 700 RPM one with a higher slope than the other two. The starting point of these curves is 40 to 45°C below the boiling point of ethanol at atmospheric pressure. In the case of Freon 113 there is an increase in performance with increasing RPM. The starting point of these curves lies 25°C below the boiling point of freon at atmospheric pressure. In general it can be said, that for the 2800 RPM the significant increase in performance starts at 40 to 45°C below the atmospheric boiling point of the working fluid.

The reason for that is, that below this value of significant increase, the rate of vapor flow inside the tube is much lower due to the retarding effects of the increased condensate flow. This is due to the increased pumping action at 2800 RPM which results from the higher pitch and the higher fins than in the case of the spiral Noranda tube. The increased pumping action drives not only the condensate

but also the vapor back to the evaporator. As the region of 40-45$^{o}$C below the boiling point of the working fluid at atmospheric pressure is passed, the vapor starts to use the whole length of the condenser as heat transfer area and the significant increase in performance is the result.

In comparison with the smooth cylinder this tube has an increase in performance of 60% at 700 and 1400 RPM and a decrease in performance of 80% at 2800 RPM for distilled water. For ethanol the increase in performance is 100% for all RPM. The Freon 113 shows an increase of 200% at 700 RPM, 270% at 1400 RPM and 300% at 2800 RPM. The higher increase for Freon 113 than for ethanol and distilled water has the same reason as discussed in Section IV.C.

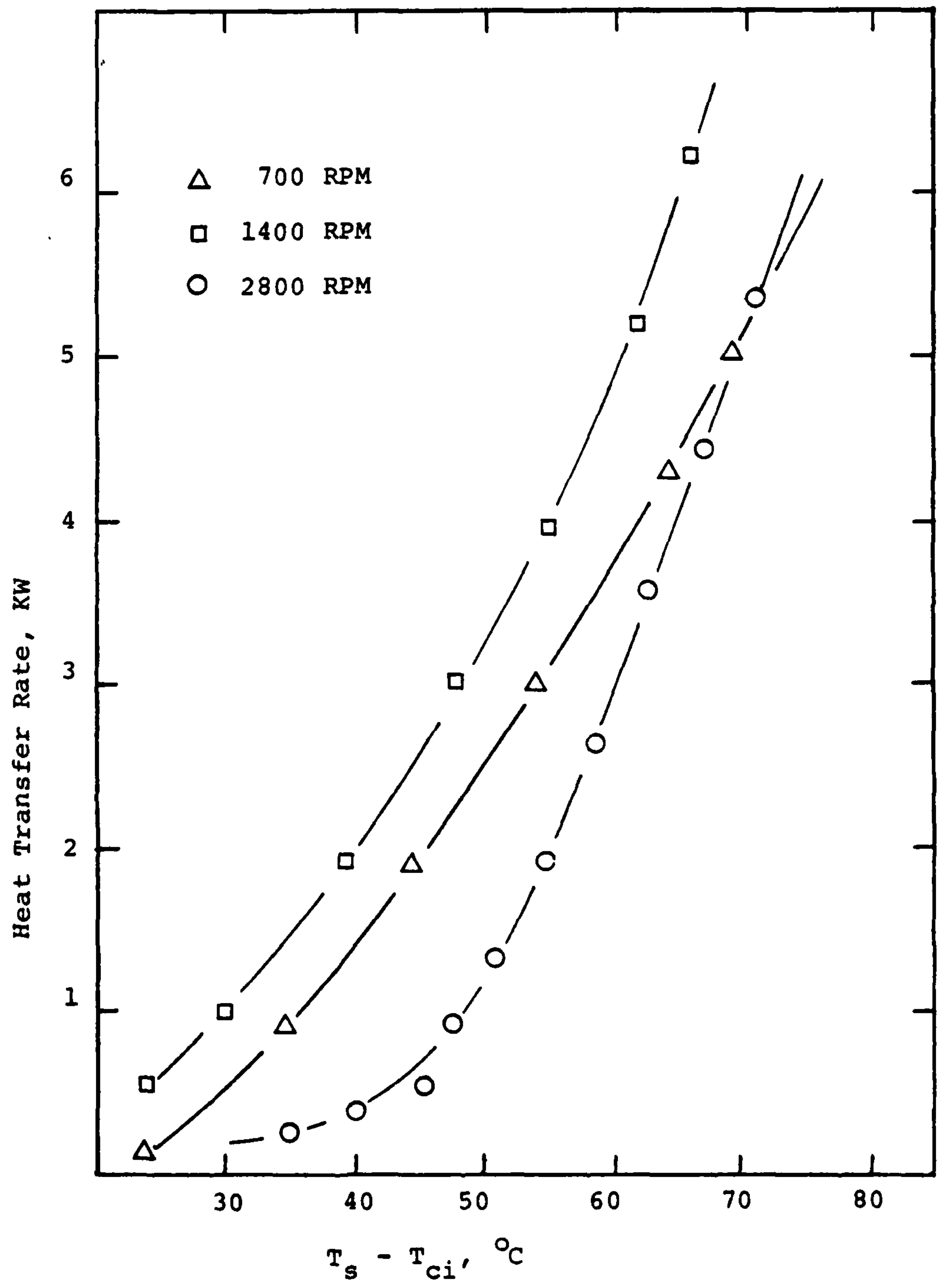



Figure 22. Heat Transfer Rate Versus Saturation Temperature Minus Water Inlet Temperature for Turbotec Tube and Distilled Water.

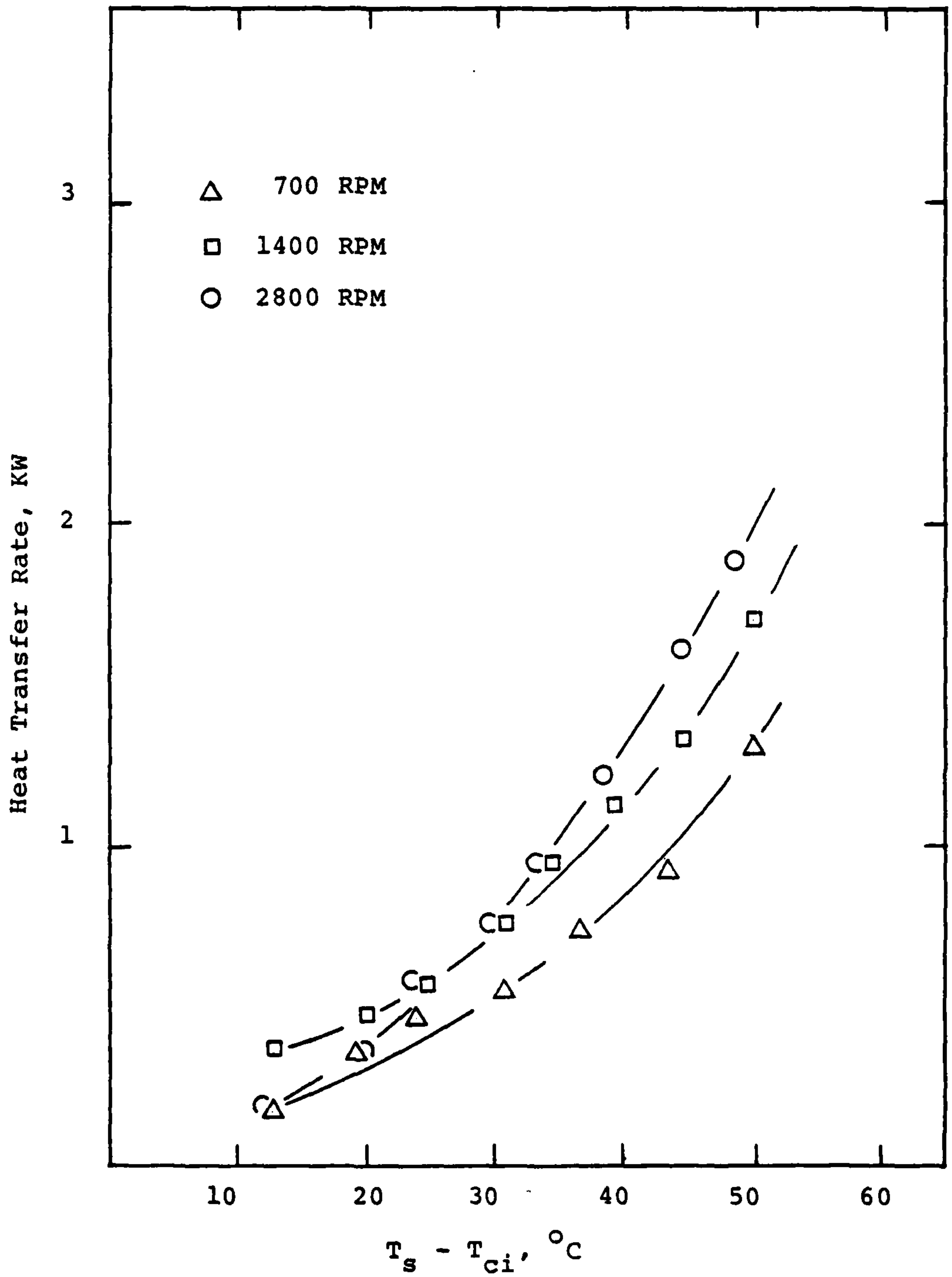



Figure 23. Heat Transfer Rate Versus Saturation Temperature Minus Water Inlet Temperature for Turbotec Tube and Ethanol.

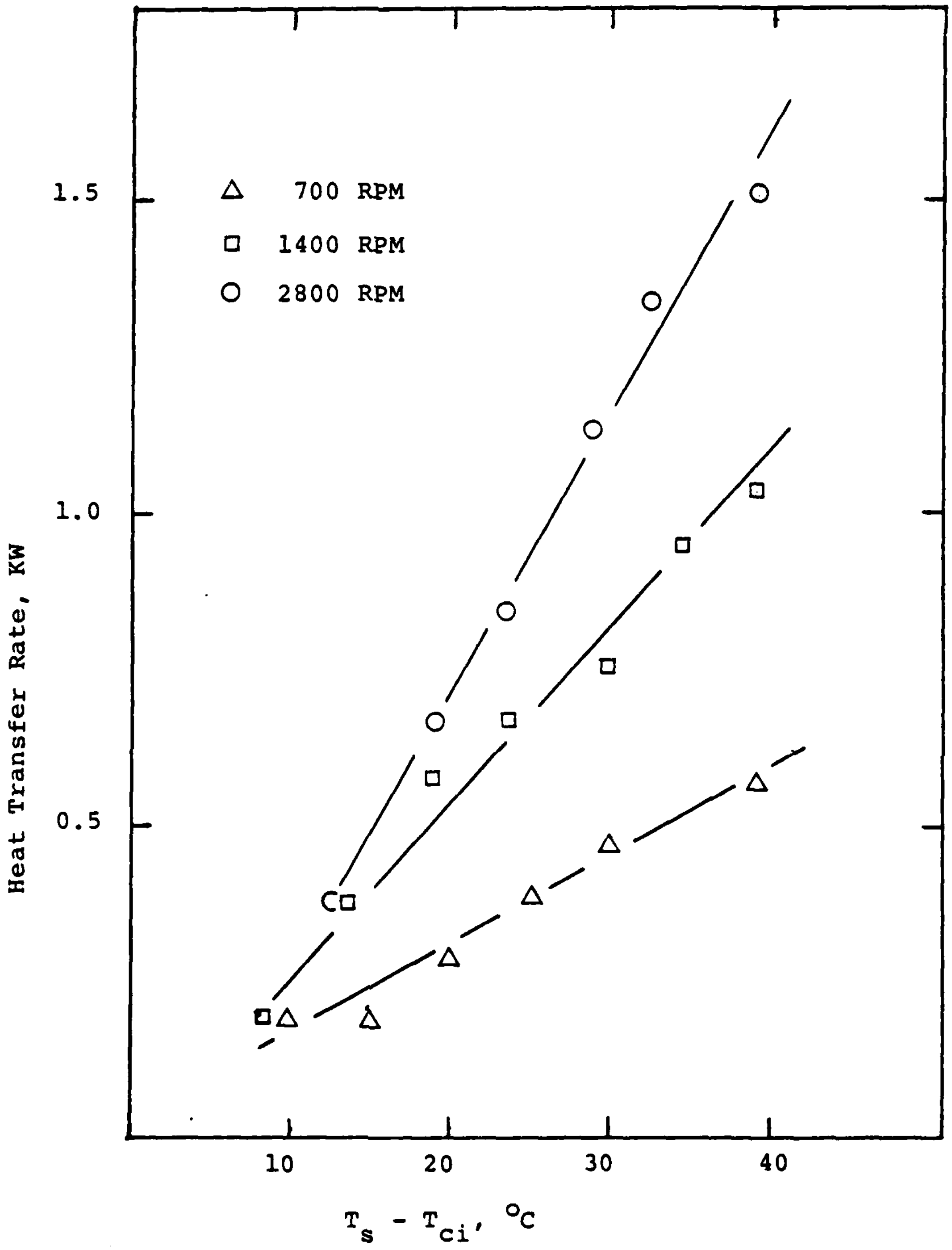



Figure 24. Heat Transfer Rate Versus Saturation Temperature Minus Water Inlet Temperature for Turbotec Tube and Freon 113.

# V. CONCLUSION AND RECOMMENDATION

## A. CONCLUSION

The following conclusions can be made based on the experimental results.

1. The best performance was achieved by using the spiral Noranda tube. The percent increase in performance over the smooth cylinder with distilled water as working fluid was 100%. The increase for ethanol and Freon 113 was even higher, up to 290% for Freon 113 at 2800 RPM at a temperature difference ($T_s$ - $T_{ci}$) of 40$^\circ$C.

2. The second best performance was given by using the Turbotec tube. This tube shows a smaller increase in performance (compared with the spiral Noranda tube) over the smooth cylinder at lower saturation temperatures, but a significant increase at higher saturation temperatures, even exceeding the performance values of the spiral Noranda tube in some points.

3. The smooth cylinder, the Hitachi Thermoexcel-C tube and the Hitachi Thermofin Tube Type II showed moderate results only by using distilled water as working fluid. The results with ethanol and Freon 113 were extremely poor especially with the Hitachi Thermoexcel-C tube. In this case, a decrease in performance over the smooth cylinder of 30 to 50% was the result.

4. The choice of the working fluid highly depends on the actual machine temperature. At lower temperatures

(about 50$^{\circ}$C), the choice of all three working fluids seems possible, because the heat transfer achieved for each fluid lies in the same range. At higher machine temperatures the use of distilled water seems to be the only choice due to the much better performance.

## B. RECOMMENDATION

Further research might be done in several areas:

1. Experimentally test spiral Noranda tubes which have different spiral angles (pitch), different number of fins, and different fin heights in order to find the best performance.

2. Also, the use of different Turbotec tubes in regard to pitch, number of fins, and height of fins should be tried.

3. Evaluate the experimental data to determine if the data can be correlated by heat pipe geometry, working fluid properties, etc.

Recommendations 1 and 2 should give a better understanding with the influence of parameters like pitch and fin configuration with respect to the heat transfer performance achieved. The goal for all further research should be to find a combination of heat pipe and working fluid which optimizes the performance in heat transfer to be competitive in real world applications.

## APPENDIX A

### Uncertainty Analysis

The uncertainty analysis of the experimental heat transfer rates was done by the method of Kline and McClintock [4].

$$Q_{total} = Q_t = \dot{m}c_p \Delta T_t$$

$$Q_{frictional\ loss} = Q_f = \dot{m}c_p \Delta T_f$$

$$Q_{corrected} = Q_t - Q_f$$

where: $Q_t$ = total heat transferred to cooling water, KW

$Q_f$ = heat transferred due to friction, KW

$\dot{m}$ = mass rate of flow of cooling water, kg/sec

$c_p$ = specific heat of cooling water, KJ/kg-$^{o}$C

$\Delta T_t$ = difference of cooling water outlet and inlet temperatures due to total heat transferred, $^{o}$C

$\Delta T_f$ = difference in cooling water outlet and inlet due to heat of friction, $^{o}$C

The uncertainties of these quantities are designated by:

$$W_Q \quad , \quad W_{\dot{m}} \quad , \quad W_{c_p}$$

The fractional uncertainties are given by:

$$\frac{W_{Q_f}}{Q_f} = \left[ \left(\frac{W_{\dot{m}}}{\dot{m}}\right)^2 + \left(\frac{W_{c_p}}{c_p}\right)^2 + \left(\frac{W_{\Delta T_f}}{\Delta T_f}\right)^2 \right]^{\frac{1}{2}}$$

$$\frac{W_{Q_t}}{Q_t} = \left[ \left(\frac{W_{\dot{m}}}{\dot{m}}\right)^2 + \left(\frac{W_{c_p}}{c_p}\right)^2 + \left(\frac{W_{\Delta T_t}}{\Delta T_t}\right)^2 \right]^{\frac{1}{2}}$$

The uncertainty for the corrected heat transfer rate is given by

$$W_{Q_c} = (W_{Q_t}^{2} + W_{Q_f}^{2})^{1/2}$$

The values and uncertainties of the measured quantities were taken to be

$$c_p = 4.187 \text{ KJ/kg-}^{\circ}\text{C}$$

$$W_{c_p} = \pm 0.0001 \text{ KJ/kg-}^{\circ}\text{C}$$

$$\dot{m} = 50\% \text{ of maximum flow rate} = 0.225 \text{ kg/sec}$$

$$W_{\dot{m}} = \pm 1 \text{ percent of } \dot{m}$$

$$W_{T_{in}}, W_{T_{out}} = \pm 0.1^{\circ}\text{C}$$

$$W_{\Delta T} = (W_{T_{in}}^{2} + W_{T_{out}}^{2})^{1/2}$$

For the smooth cylinder condenser at 700 RPM and distilled water as working fluid, the uncertainty analysis is given as

$$Q_t = 1.882 \text{ KW}$$

$$Q_f = 0.188 \text{ KW}$$

$$\Delta T_t = 1.1^{\circ}\text{C}$$

$$\Delta T_f = 0.2^{\circ}\text{C}$$

$$\dot{m} = 50\% \text{ maximum flow rate } (0.225 \text{ kg/sec})$$

$$c_p = 4.187 \text{ KJ/kg-}^{\circ}\text{C}$$

and

$$W_{Q_f} = Q_f \left[ \left(\frac{W_{\dot{m}}}{\dot{m}}\right)^2 + \left(\frac{W_{c_p}}{c_p}\right)^2 + \left(\frac{W_{\Delta T_f}}{\Delta T_f}\right)^2 \right]^{1/2}$$

$$= .188 \left[ \left(\frac{1}{50}\right)^2 + \left(\frac{0.0001}{4.187}\right)^2 + \left(\frac{0.1}{0.2}\right)^2 \right]^{1/2}$$

$$= .188 \quad (.5)$$

$$= \pm 0.094 \text{ KW}$$

$$Q_f = 0.188 \pm 0.094 \text{ KW}$$

$$W_{Q_t} = Q_t \left[ \left(\frac{W_{\dot{m}}}{\dot{m}}\right)^2 + \left(\frac{W_{c_p}}{c_p}\right)^2 + \left(\frac{W_{\Delta T_t}}{\Delta T_t}\right)^2 \right]^{1/2}$$

$$= 1.882 \left[ \left(\frac{1}{50}\right)^2 + \left(\frac{0.0001}{4.187}\right)^2 + \left(\frac{0.1}{1.1}\right)^2 \right]^{1/2}$$

$$= \pm 0.175 \text{ KW}$$

$$W_{Q_c} = \left[ (W_{Q_f})^2 + (W_{Q_t})^2 \right]^{1/2}$$

$$= \left[ (0.094)^2 + (0.175)^2 \right]^{1/2}$$

$$= \pm 0.199 \text{ KW}$$

$$Q_c = (1.882 - 0.188) \pm 0.199$$

$$Q_c = 1.694 \pm 0.199 \text{ KW}$$

Uncertainties in the heat transfer rate are plotted on Figures 10, 11, 12, and represent typical performance uncertainties.

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